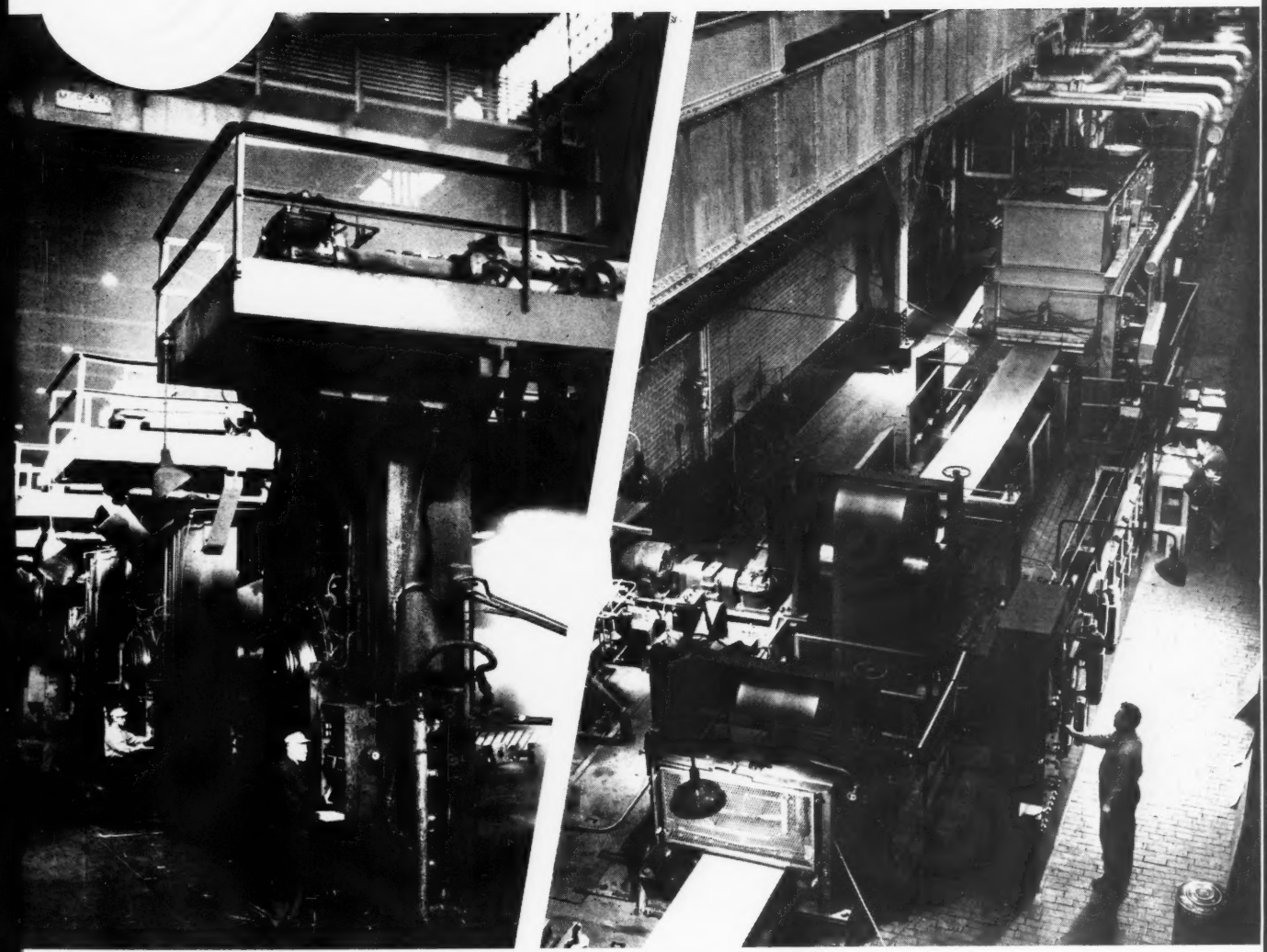
September-October, 1954
Vol. 10, No. 5

Lubrication Engineering



Plant Lubrication Program (p. 249)

JOURNAL OF
THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

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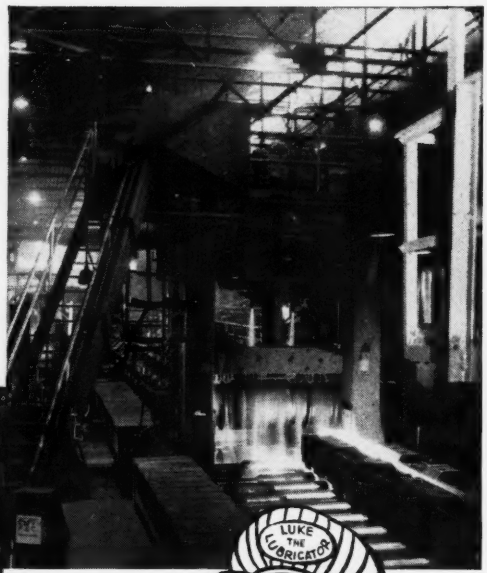
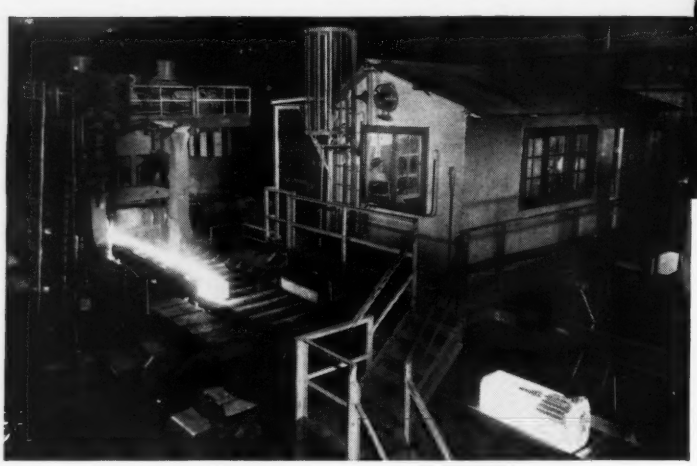
.....and cuts MAINTENANCE *to the bone!*





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Membership in the American Society of Lubrication Engineers is in several grades as defined below. Assignment to grade is made by the Admissions Committee or Board of Directors on the basis of information submitted or supplied by references.

MEMBERS: Members shall be persons not less than 24 years of age who: (1) are engaged in research and instruction at technical schools, universities, and various publicly and privately supported institutions in the field of lubrication; or (2) have occupied recognized positions as lubrication engineers for a period of three or more (not necessarily consecutive) years prior to date of admission, involving the responsibility for or supervision of the development, selection, field use and application of lubricants as differentiated from other activities; or (3) are indirectly concerned with the field of lubrication, but possessing other qualifications of experience, knowledge, and accomplishment, and have manifested a particular interest in the purposes and welfare of the Society to the extent that their membership would be a valuable contribution to the successful functioning of its activities. Fee $12.50.

ASSOCIATE MEMBERS: Associate Members shall be persons less than 24 years of age, and those who do not completely fulfill the membership requirements for Members. Fee $6.25.

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For application blanks or further information, write:

ASLE
84 E. Randolph St.
Chicago 1, Ill.

September-October, 1954
Vol. 10, No. 5

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(Cover Pictures courtesy of U. S. Steel Corp.)

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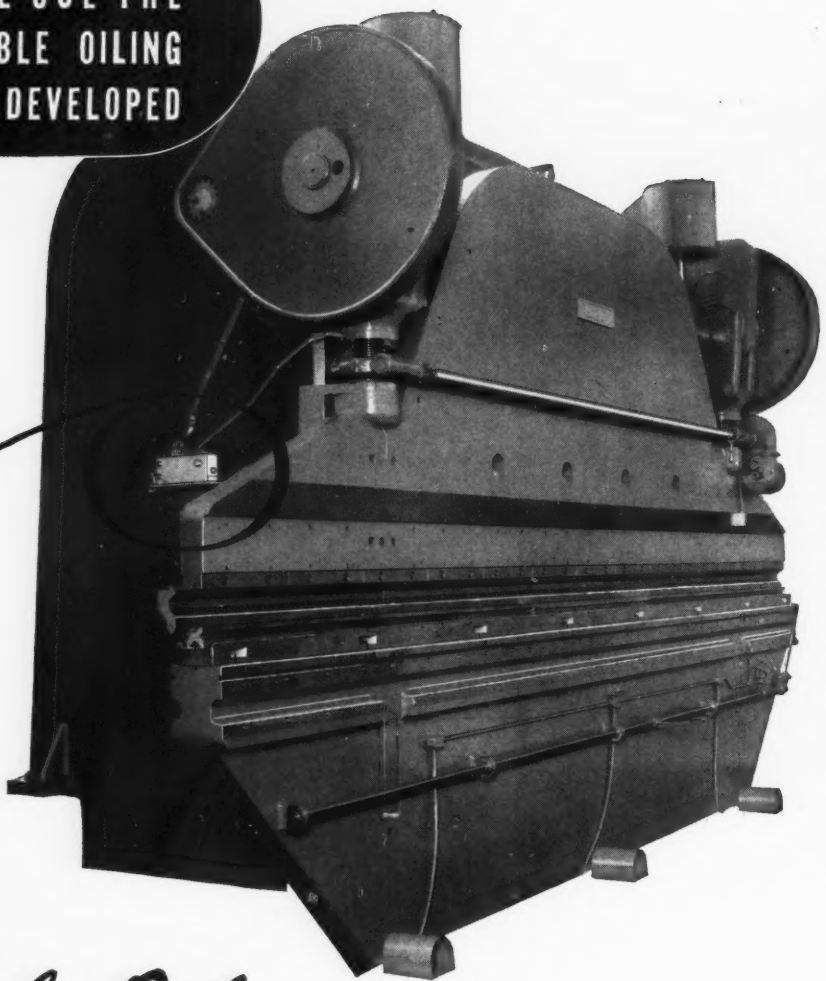

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Meet the Authors

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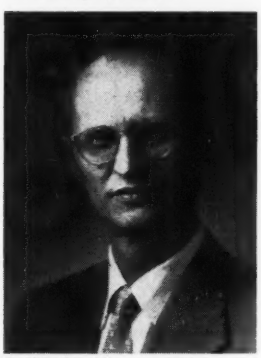

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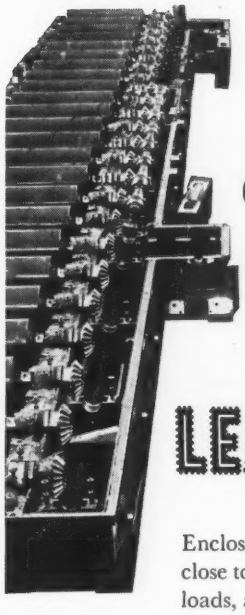

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NEW 'LUBE LINES' COORDINATOR. In our efforts to make Lube Lines more informative, the job of coordinating information for it has been given to Mr. Erwin Landau, Asiatic Petroleum Corp., 655 Fifth Ave., New York 22, N. Y. All readers who have any new developments or personnel changes they would like to publicize, should send the details to him at the above address. S.K.T.

ASLE PITTSBURGH SECTION TO SPONSOR CONFERENCE. The ASLE Pittsburgh Section is sponsoring a Conference on Lubrication, November 10 & 11, Pittsburgh, similar to the one cosponsored with the University of Pittsburgh several years ago. This year's theme will stress the "selection of lubricants" from an applied rather than a theoretical viewpoint, with a tentative program as follows: November 10, Hydraulic Fluids, Cutting Fluids, and Metal Working Lubricants; November 11, Gear Lubricants, Greases, Machine Oils, and Internal Combustion Engine Lubricants. For complete details, write: S. S. McNary (Secretary), 87 Woodlawn Ave., Crafton, Pittsburgh 5, Pa.

M.I.T. SUMMER SESSION HI-LITES. Thirty-two students attended the ASLE sponsored Lubrication Engineering Course at the Massachusetts Institute of Technology, Cambridge, Massachusetts, June 15-25, 1954, under the supervision of Prof. B. G. Rightmire (in charge of the Lubrication Laboratory in the M.I.T. Mechanical Engineering Department.)

Pictured above, from left to right, are (1st row): J. F. Wickey, W. C. Brandow, J. E. Hogstad, K. E. Snow, P. H. Bowen, A. E. Rahm, R. B. Prizzi; (2nd row) B. I. Patterson, C. T. Stone, C. A. Chilson, P. Levi, J. N. Hocke, J. W. Baybutt, D. J. Pritchard, J. J. Lamb, G. G. Bull; (3rd row) R. G. Rieser, L. B. Sheetz, A. H. Pienkoski, J. D. Neesley, J. V. Kalb, E. F. Ryan, R. P. Kerevel, P. C. Davis, B. G. Rightmire; (4th row) K. A. Arnold, G. E. Crouch, R. B. Lynch, A. G. Finney, D. J. Meier, T. L. Giard, F. R. Lydecker, R. C. Cuny.

PLAIN TALK, by L. C. Beard, Jr., Socony-Vacuum Oil Co., Inc., (Reprinted in part from Annual Address by the President, American Society for Testing Materials.)

"I shall take advantage of the opportunity afforded each President to make an annual address, and shall discuss an unhealthy trend in science which is giving much concern and about which we all can do something: the breakdown in communications between the various sciences and between science and the public.

I say to you: 'La scienza e scienza solamente quando il primo spazzaturaio ti comprende,' and unless you understand Italian and can surmount my mispronunciation, this means nothing to you. If I translate this old proverb: 'Science is science only when the first streetcleaner understands you,' you know what I'm talking about and you also have the text of this address. I could call the salad you have just eaten: 'Lamellar fibrous cellulose impregnated with chlorophyll and tumbled with a triglyceride-acetic acid emulsion.' Frankly, I prefer to call it: 'green salad with French dressing.'

The sciences have become so complex and workers in them have built up their own vocabularies to such an extent that a chemist has difficulty in understanding a physicist; the biologist can't talk to the geologist; none of them can talk plainly to the engineer. Some times the engineer can't get his ideas across to management; and few technical people can explain their aims and what they have done, to the public.

Vannevar Bush, addressing the Medical Society of the District of Columbia last year, warned that over specialization and the volume of publication are driving groups of scientists to develop their own jargon, unintelligible except to the initiated, heightening the barriers which separate their works from the main stream of progress. He said there were two concrete things that could be done about it. First, scientists could be intelligible. He said they have a moral obligation to express themselves so that they can be quickly understood by those within their profession or closely related professions. Second, those who possess that unique skill should be encouraged to gather, integrate, and interprete the information developed by many workers so that great masses of seemingly confused data become clear, intelligible, and useful to their fellows.

Melba Phillips reporting for a group of scientists in the magazine Science said: 'In England, France, and Germany, there sprang up in the nineteenth century a tradition of the scientist's obligation to interpret his philosophy and his results to the nonscientific public to which, in the broadest sense, he owed his support. At the present, Albert Einstein is an outstanding exemplar of this tradition, which was notably supported in the past by Faraday, Pasteur, Huxley, and Poincare, among others. Most American scientists have retreated so far into the remote reaches of specialization that they have largely forgotten their obligation to the public.'

(Continued on page 281)

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You can't improve on the ORIGINAL, PROVED lithium-base multi-purpose grease by INTERNATIONAL.

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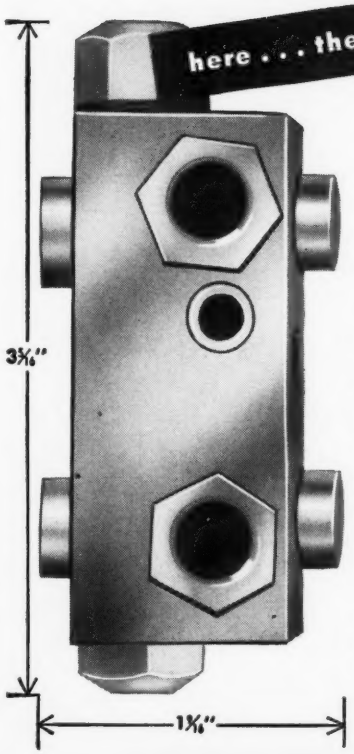

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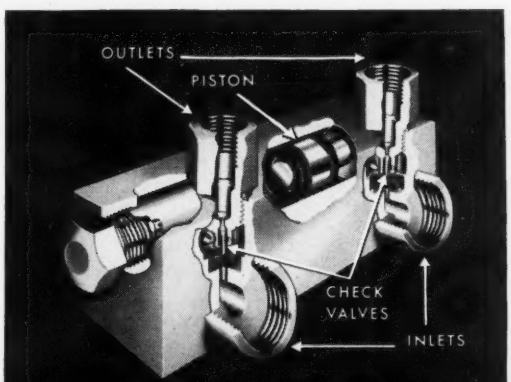

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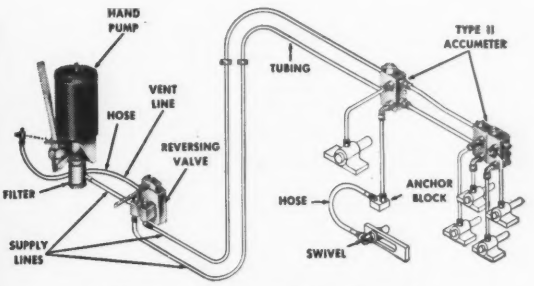

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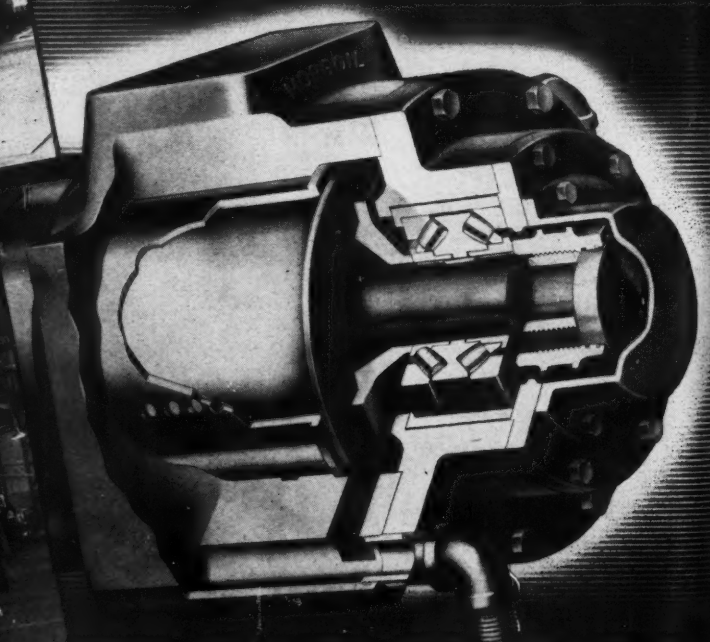

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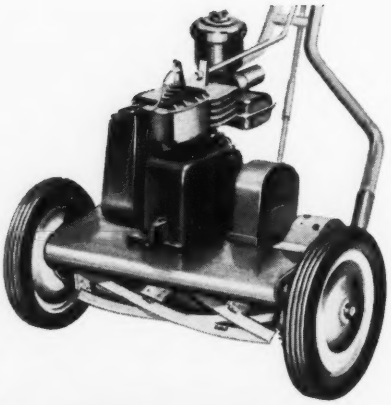

1. **They're uniform**—both in sealing contact and in spring load. KLOZURES are precision made to the highest standards of accuracy.

2. **They save space**—these seals are compact and light in weight, yet they're rugged.

3. **They give positive bearing protection**—KLOZURES seal the lubricant in *and* keep dust, dirt and water out.

4. **They save money**—made in radial widths smaller than conventional oil seals and with fewer parts.

Model 93-B—A narrow cross-section seal with synthetic rubber outer covering for soft metal housings.

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Model 71-B—A narrow cross-section spring-loaded seal for either light or heavy lubricants.

Leading manufacturers of power lawn mowers and power lawn mower engines know they can depend on Garlock KLOZURE oil seals for maximum bearing protection under rugged service conditions. Engine manufacturers use KLOZURES on the output shaft. Power lawn mower makers use Garlock KLOZURES to protect bearings on wheels, reels and gear boxes.

Let us show you how Garlock KLOZURE oil seals can prolong the life of your bearings. There's a service-tested KLOZURE model for every bearing application. Get all the facts —call your GARLOCK representative or write for KLOZURE Catalog No. 10.

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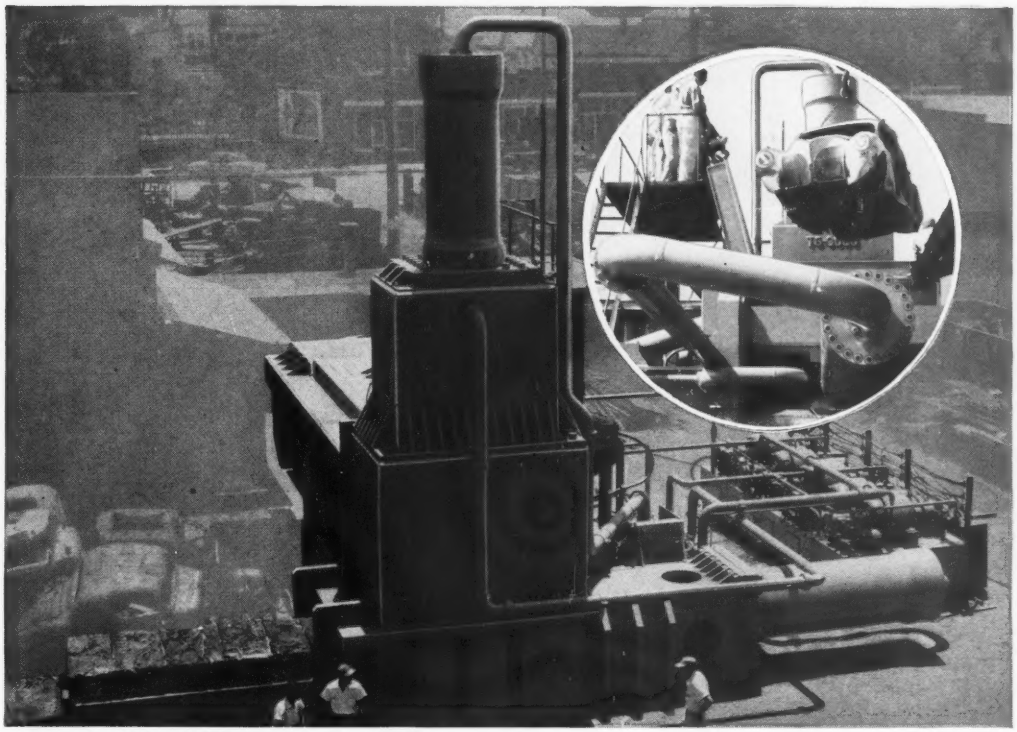
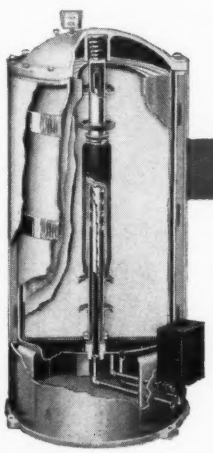


An outstanding example of industry's modern, push-button miracles is the new TG-3000, gigantic scrapmetal baling press specially built by the Harris Foundry & Machine Company for the Chicago Auto Wrecking Corporation.

Weighing 675,000 pounds, the huge, 800-horsepower press is capable of baling up to 75 tons per hour. It actually can compress two or more complete autos simultaneously into neat packages measuring about 24″ x 24″ x 60″ each.



HONAN-CRANE OIL PURIFIER PROTECTS 3700 GALLON HYDRAULIC SYSTEM

The TG-3000 requires a hydraulic system containing 3700 gallons of high grade hydraulic oil. Push-button controls connected to the hydraulic circuit make the press manual, semi-automatic or fully automatic in operation.

To insure against costly breakdowns in this complex system, the Harris Company will equip it with a Honan-Crane Bulk-Type Oil Purifier as shown at left. Honan-Crane purification protects vital hydraulic mechanisms, extends safe-use life of oil, insures maximum operating efficiency and economy.

Give *your* plant the benefit of Honan-Crane's 30 years' filtration experience. There's a Honan-Crane Purifier for complete removal of either solid or soluble contaminants from hydraulic, lubricating, metal-working, insulating and other oils.

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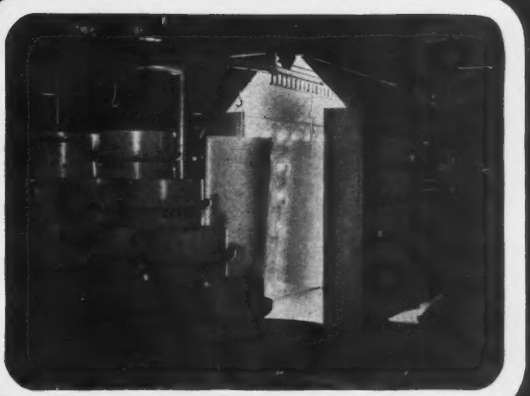

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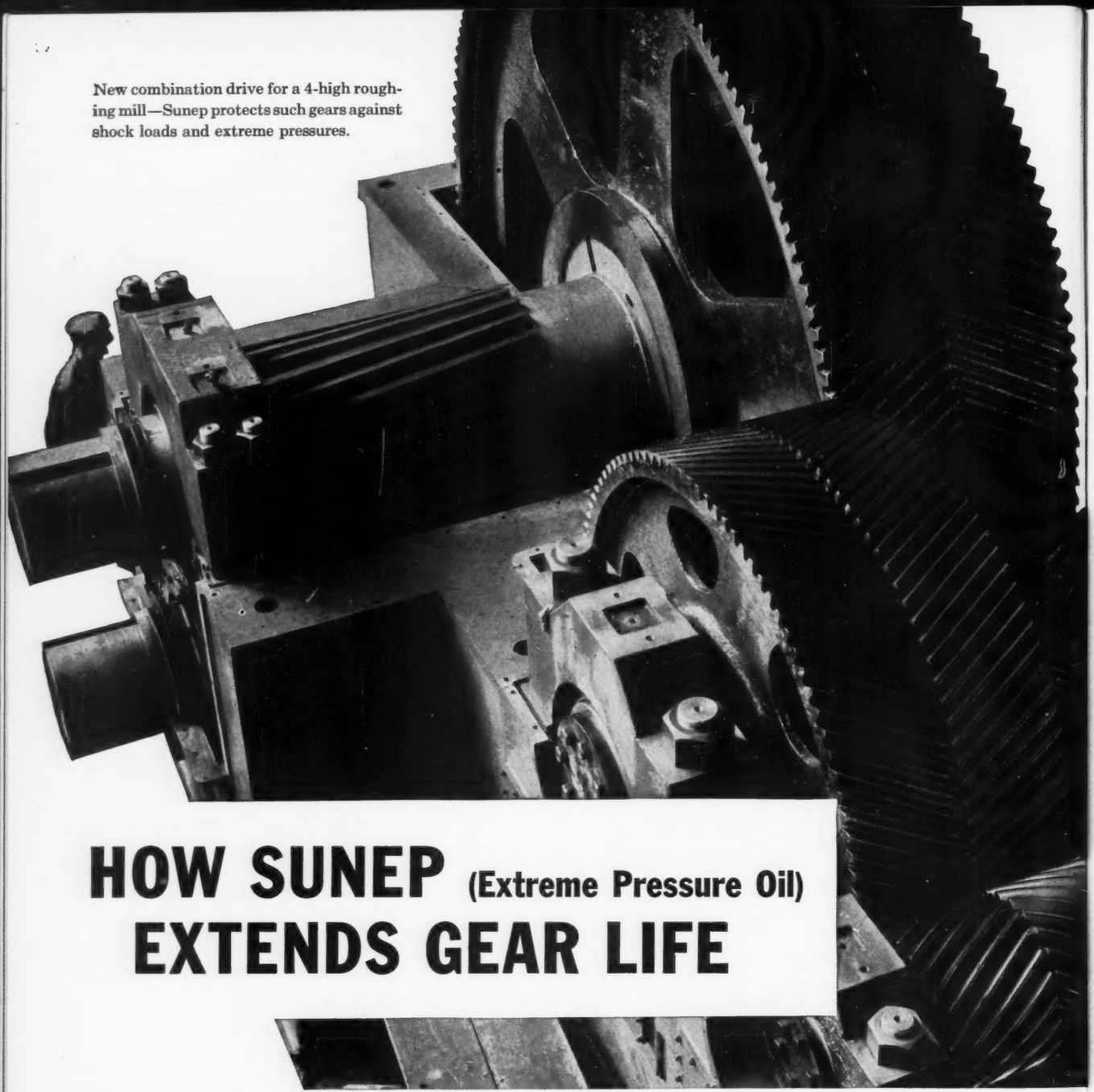

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ASME-ASLE Lubrication Conference

October 18 & 19, 1954

Lord Baltimore Hotel, Baltimore, Md.

PROGRAM

Session I, Hydrostatic Bearings. Oct. 18, 10:00 a.m., D. D. Fuller, Chrmn.

An Analysis of Orifice Compensated Hydrostatic Journal Bearings, by A. A. Raimondi & J. Boyd

Behavior of Air in the Hydrostatic Lubrication of Loaded Spherical Bearings, by T. L. Corey, H. H. Rowand, Jr., E. M. Kipp & C. M. Tyler, Jr.

Session II, Hydrodynamic Bearing Analysis. 1:30 p.m., D. F. Wilcock, Chrmn.

Recent Advances in the Hydrodynamic Theory of Slider-Bearing Lubrication, by F. Osterle & E. Saibel

On the Solution of Reynolds' Equation for Slider Bearing Lubrication, IX—The Stepped Slider with Adiabatic Lubricant Flow, by F. Osterle, A. Charnes & E. Saibel

Studies in Lubrication, X—The Complete Journal Bearing with Circumferential Oil Inlet, by M. J. Jacobson, A. Charnes & E. Saibel

The Short Bearing Approximation for Plain Journal Bearings, by G. B. DuBoise & F. W. Ocvirk

Power Loss in Elliptical and 3-Lobe Bearings, by O. Pinkus

Session III, Bearing Instability. 8:00 p.m., F. C. Linn, Chrmn.

Investigation of Translatory Fluid Whirl in Vertical Machines, by B. Sternlight & G. F. Oeker

Oil Film Whirl—An Investigation of Disturbances to Oil Films in Journal Bearings, by B. L. Newkirk & J. F. Lewis

Turbulence in Water-Lubricated Thrust Bearings, by S. Abramovitz

Session IV, Rolling Element Bearings & Cams. Oct. 19, 9:30 a.m., E. E. Bisson, Chrmn.

Materials & Designs of Gages for High-Speed RollerBearings, by W. J. Anderson & Z. N. Nemeth

Hydrodynamic Lubrication of a Cam & Cam Follower, by R. Davies

A Progress Report on the Surface Endurance Limits of Engineering Materials, by G. J. Talbourdet

Session V, Friction & Lubricants. 1:30 p.m., W. E. Campbell, Chrmn.

Friction in a Close-Contact System, by W. Claypoole

The Statistical Nature of Friction, by E. Rabinowicz, B. G. Rightmire, C. E. Tedholm & R. C. Williams

An Analysis of Recent Data on the Effects of Temperature & Pressure on the Viscosity of Lubricants, I—Paraffinnic & Napthenic Base Oils, by R. B. Dow

Some Experiments on the Flow of Oil & Grease through Tubes of Circular Cross Section, by H. E. Mahncke & W. Tabor

ASLE Annual Meeting and Exhibit

April 13-14-15, 1955

Hotel Sherman, Chicago, Ill.

Forty-one papers have been tentatively scheduled in 16 sessions for the 10th Annual Meeting & Lubrication Exhibit of the American Society of Lubrication Engineers; five of these sessions will be devoted exclusively to a special short course in Lubrication Engineering during the meeting.

One of the hi-lites on the Program will be a Panel on Industrial Hygiene & Material Hazards which will discuss dermatitis and its prevention; another Panel will cover "Controls for Extending the Useful Life of Coolants While in Use."

Two Symposia will be held during the Meeting, the first a Symposium on "Fundamentals of Wear" which will cover the latest theories and developments in this research field; the second will cover "Plant Safety as Related to Lubrication" with discussion on non-flammable hydraulic fluids, advantages of centralized lubrication, proper dispensing equipment, and safeguards against injury and fire.

Other sessions will be devoted to: Film type bearings, Properties of lubricants, Present thinking in gear lubrication, Lubricant dispensing, Equipment and application, Rolling contact bearings, Organization of a plant lubrication program, Metal working fluids, Lubricating greases, and Hydraulics.

PLANT LUBRICATION PROGRAM

by J. F. Boal*

This paper presents a lubrication program which was developed to fit the exacting needs of an integrated steel plant. It contains many procedures which are directly applicable to any manufacturing plant.

In considering the lubrication program in a plant it is first well to get a general idea of the manufacturing operations carried on in the plant, the type of machinery used in the operations and something of the type of organizational structure in the plant.

The typical integrated steel plant consists of a number of operating units or divisions each of which carries out a part of the process of converting iron ore into a variety of semi-finished and finished steel products such as slabs, blooms, billets, bars, plates, strip, sheets, tin plate, structural shapes, pipe and others. The principal operating divisions usually include coke ovens, blast furnaces, open-hearth furnaces, primary hot-rolling mills, hot-finishing mills, cold-reduction mills and sheet- and tin-finishing mills. In addition to the operating divisions, the steel plant has a number of service divisions the activities of which are plant wide. These include Central Maintenance, Power and Fuel, Transportation, Metallurgical, Engineering, Production Planning, Industrial Relations and Accounting.

There is a great variety of both stationary and movable equipment in a steel plant. The machinery is ruggedly constructed, not only because of the heavy loads and severe operating conditions, but also because it is designed for continuous operation over long periods of time without breakdown.

The need for protecting the operating equipment against excessive wear and for keeping mill delays to the absolute minimum brings lubrication into a very prominent position in the steel plants' operation. Every bearing, gear, and other moving part of the equipment requires lubrication to permit it to function successfully and contribute its part toward high plant production. For example, a bearing failure on a pinion stand in a rolling mill can cause a delay of several hours in that mill and, since there is a continuous flow of steel through the plant, it may seriously interfere with the scheduled operation of the whole plant including the open hearth department which supplies steel to the rolling mill. In such a situation, many dollars and tons of production can be lost as a direct result of a lubrication failure.

The success of any plant-wide lubrication program depends upon the close cooperation among certain individuals and groups of individuals within the various units and divisions of the organizational structure, including the following:

Plant Lubrication Engineer. He is usually a member of the staff of the Central Maintenance Superintendent, which staff initiates and supervises maintenance programs on a plant-wide basis. He will not be in direct charge of lubricant application, but rather will collaborate with the supervisory and lubrication personnel in each operating unit to create an effective plant-wide lubrication program.

Lubrication Groups in each operating unit. These groups are directly responsible for the application of lubricants and for the installation and maintenance of lubrication equipment within their respective operating units. The personnel in these groups may have special training for work of the type.

Equipment operators. In some instances men who operate individual, isolated machines or equipment are responsible for their lubrication. These men should receive thorough instructions in the proper lubrication of the equipment for which they are responsible.

Lubricants Storekeeper. He places orders with the lubricant suppliers, receives and stores lubricants in the central storehouse, keeps the operating divisions supplied with lubricants and provides data for reports on consumption of lubricants.

Engineering Department. They play an important role in plant lubrication through their design and specification for new equipment.

Industrial Engineers. They assist in obtaining approval for new lubrication equipment.

Plant Director of Training. He and his staff render valuable assistance in setting up and administering training programs in lubrication for all levels of plant lubrication personnel.

PLANT-WIDE LUBRICATION PROGRAM. Prior to initiating a plant-wide lubrication program, the Plant Lubrication Engineer should prepare an outline to guide his own activities as well as to show others what he plans to do. A suggested outline which contains all of the factors recognized as being essential for conducting a successful plant lubrication program is shown in Figure I.

*United States Steel Corporation, Clairton, Pa.

This paper was presented at the 5th Session of a Lubrication Engineering Course sponsored by the New York Section of ASLE, New York, N. Y., March 11, 1954.

This, then, will be the plant program. After referring it to top management for approval, it is advisable to present it to the superintendents of the various operating divisions for their comments and suggestions. The latter step is important in obtaining wide acceptance of the program by giving all groups in the plant an opportunity to participate in its formulation and initial steps.

At this point the importance of human relations in lubrication engineering should be stressed as this can contribute much to the success or failure of any program undertaken. In performing his job the Lubrication Engineer must deal with various persons and groups of persons in different levels throughout the plant. It is essential that he be able to cooperate with others to a rather high degree. Ideas or programs have to be sold. In order to obtain ready acceptance, they must be presented in a clear logical manner and be supported by the necessary justifying data and information. Four characteristics are regarded as essential for success in most types of work similar to that of plant Lubrication Engineering. They are: (1) Ability to get along with people. (2) Ability to sell yourself and your ideas. (3) Technical competence. (4) Initiative to apply the above three and get things done.

LUBRICATION SURVEY OF THE PLANT. If the plant has not recently been surveyed from a lubrication standpoint, a careful survey should be conducted to determine the current lubrication practices. The storeroom will furnish a list of the name brand lubricants coming into the plant and the monthly or yearly consumption of each. The method of receiving and storing these lubricants should be established.

The main part of this survey will consist of a detailed lubrication inspection of all plant equipment. The Lubrication Engineer should work with one operating unit at a time, and he will require the assistance of the maintenance foreman, the lubrication men, and perhaps the millwrights in the operating unit who are thoroughly familiar with the equipment, the lubricants being used and any special maintenance problems in connection with the equipment.

Each machine must be studied and its various lubrication points listed by name such as: electric-motor bearings, flexible couplings, table reduction gears and bearings, table lineshaft bearings, and the like. The brand name of the lubricant being used, together with the means and frequency of application also should be listed for each lubrication point. Likewise any abnormal conditions encountered such as excessive oil leakage, water entering the oil, frequent bearing changes and the like should be recorded.

Obtaining this data is time consuming and several months may be required to survey the plant. However such a survey is the only way of obtaining an accurate picture of current lubrication practices and it is the basis upon which future steps to improve and systematize lubrication will be made. Therefore the time spent in conducting a thorough survey is well justified. Since a general knowledge of the design of a machine is a requisite for making any critical analyses of its lubrication requirements, it may be necessary for the Lubrication Engineer to make frequent reference to the blueprints to clarify points that cannot be determined by an external inspection. Where particular lubrication or other maintenance problems are being encountered on a machine every effort should be made to inspect the machine when it is opened for repair to determine the particular points where excessive wear or other trouble is taking place.

After the survey of each operating unit has been completed, a summary report of the findings should be presented to the Maintenance Superintendent in that unit. This report should indicate any bad conditions which eventually may be improved by changes in practices. The details of such changes may be deferred until sufficient time is available to develop them.

CONSOLIDATION OF BRAND-NAME LUBRICANTS UNDER DEFINITE TYPES. From an engineering standpoint, any work in lubricant application should be based upon lubricant types rather than simply upon the supplier's brand names of the lubricants. While it requires some study of lubricant characteristics to establish lubricant types, this is important, as it gives a flexibility to the pro-

Plant Lubrication Program

1. SURVEY LUBRICATION NEEDS.
2. CLASSIFY LUBRICANTS BY TYPES.
3. PREPARE LUBRICATION CHARTS.
4. EVALUATE NEW LUBRICANTS.
5. IMPROVE APPLICATION METHODS.
6. IMPROVE HANDLING AND STORAGE.
7. ESTABLISH CONSUMPTION REPORTS.
8. SET UP OIL MAINTENANCE METHODS.
9. COLLABORATE ON EQUIPMENT DESIGN.
10. INVESTIGATE SPECIAL PROBLEMS.
11. COLLABORATE IN TRAINING PERSONNEL.

Fig. 1 Plant Lubrication Program.

gram that is not possible when adhering strictly to brand names.

The Lubrication Engineer should establish a list of lubricant types which is sufficiently extensive to cover essentially all the lubricant requirements for the entire plant. Each type should have characteristics which makes it particularly suitable for applications under certain operating conditions. A common name or number should be assigned each type for use on lubrication charts, storeroom order cards, and the like.

An example of lubricant types established for a steel plant is given in Figures II and III. These show only a part of the types of oil and grease necessary to lubricate the plant. The total number of different lubricants required, including the various viscosities and consistencies, may amount to 50 or 60 for a large plant.

LUBRICANT	GENERAL CHARACTERISTICS	TYPICAL APPLICATIONS
Engine Oil - 300	Ordinary quality straight mineral oil, low pour point	Ring oiled motors Engine systems Machine tools
Circulating Oil - 1400	Medium quality mineral oil, good water separation	Back-up roll bearings
Turbine Oil - 150	Highly refined, high stability oil	Direct drive turbines Newer machine tools
Turbine Oil - 300	Highly refined, high stability oil	Geared turbines Newer machine tools
Detergent Oil - SAE 30	Good stability detergency	Diesel engines, trucks air compressors, blowing tubs
E P Oil - 110	High film strength, water separation good stability	Heavy duty enclosed gear cases, table and screwdown circulating systems

Fig. 2 Lubricant Types — Oils.

Some typical applications are noted for each type of lubricant as a general guide to be followed in consolidating brand-name lubricants throughout the plant and in preparing lubrication charts for new as well as existing equipment. The typical applications should be, as far as practical, those generally recognized by both industry and the lubricant suppliers as being among acceptable applications for the particular type of lubricant.

After the lubricant types needed for plant wide application have been established and the characteristics of all brand-name lubricants currently being used in the plant have been examined, it may be found that frequently different brand name lubricants will fall within the same type. This indicates unnecessary duplication in procurement and storage of lubricants and will make for unnecessary confusion in their handling and application. In such cases, it is generally advantageous to consolidate the entire schedule so as to buy only one brand-name lubricant under each type.

LUBRICANT	GENERAL CHARACTERISTICS	TYPICAL APPLICATIONS
EP Grease - 0	Calcium soap grease with heavy vis. oil and EP additive. Good pumpability. 200° F. dropping point.	General purpose grease for applications under 150° F. Plain and anti-friction table roller bearings. Work roll bearings.
High Temp. EP Grease - 0	Anhydrous calcium soap grease with medium vis. oil and EP additive Good pumpability. 450° F dropping point.	Plain and antifriction bearings over 150° F. Hot mill front and back tables. Coke oven door screws.
High Stability Grease - 1	Lithium or sodium grease with medium vis. oil. 350° F. dropping point.	Antifriction bearings on motors and fans.
Graphited Roll Neck Grease - 0	Calcium soap grease with heavy oil and graphite. 200° F. dropping point. Good water resistance.	Babbitt and bronze roll neck bearings.
Leaded Gear-shield - 500	Asphaltic fluid containing lead soap. Highly adhesive water resistant film.	Open and semi-enclosed gears. Plain bearings exposed to severe pressures.

Fig. 3 Lubricant Types — Greases.

The general characteristics of any particular lubricant may be determined from information received from the supplier or, if laboratory testing facilities are available, from testing a sample of the lubricant to determine its measurable physical and chemical properties. The final selection of a particular lubricant to be used should be made only after a careful observation of the lubricant in one or more typical applications in the plant.

A Plant Procurement Schedule can then be made for the benefit of the plant Storekeeper which lists the lubricant types and the brand-name lubricant currently being purchased against each type. A sample of such a schedule is shown in Figure IV.

PREPARATION OF LUBRICATION CHARTS. After the detailed plant survey has been conducted, the lubricant types established and the necessary consolidation of lubricants effected, the next step is the preparation of lubrication charts for all plant equipment. A typical lubrication chart for a piece of steel plant equipment is shown in Figure V. It consists of the name of each lubrication point or group of points requiring the same lubricant, the method of application, the type of lubricant to be used, and the frequency of application. Use of the type name instead of the supplier's brand-name results in a more permanent chart since it is then possible to change the brand-name lubricant without the chart becoming out of date. The charts should be submitted to the Maintenance Superintendent or Foreman in the Operating Unit for his approval and then posted in such a manner as to be

readily available to the men who perform the lubrication. The purpose and use of the charts should be explained to the lubrication men by their Foreman.

LUBRICANT	SUPPLIER	BRAND
Engine Oil - 150	John Doe Oil Co.	A oil
Engine Oil - 300	John Doe Oil Co.	B oil
Engine Oil - 600	John Doe Oil Co.	C oil
Circulating Oil - 1400	Richard Roe Oil Co.	X oil
Circulating Oil - 2400	Richard Roe Oil Co.	Y oil
Turbine Oil - 150	Jos. Smith Oil Co.	R oil

Date: 2-11-54

William Smith
Lubrication Eng.

Fig. 4 Plant Procurement Schedule.

Lubrication charts are important as they represent definite written instructions, prepared by the Plant Lubrication Engineer, approved by the Operating Maintenance Supervisor, and placed in the hands of the mill personnel who apply the lubricants, for their guidance. Without such charts there can be much confusion and difference of opinion in the plant as to what should be used or even what is actually being used in the different equipment, especially when there has been a change in either the maintenance supervision or the lubrication personnel in the plant. The mere issuing of the charts, of course, does not insure compliance. Periodic checks by the Foreman or Lubrication Engineer are necessary to see that correct practices are being followed.

Every effort should be made to keep to a minimum the number of different lubricants within a single operating unit. This will require some compromise from an application standpoint, but usually it is more satisfactory than attempting to use the most desirable lubricant for each bearing and gear set that the operating conditions would indicate.

It will be necessary to alter lubrication charts upon installation of new application devices, change in application frequency or change in lubricant type.

The aim, of course, is to eventually have charts which indicate the proper type of lubricant, applied by the most efficient method at the proper intervals. Obviously this cannot always be realized at the onset of a program but it should be kept in mind as the ultimate goal in lubricant application.

EVALUATE NEW LUBRICANTS. Since most lubricant manufacturers conduct continuous research to develop lubricants with more desirable combinations of characteristics, it is difficult to state that any particular lubricant is the "best" lubricant. A lubricant that may appear to be the best this year may be surpassed by a newly developed lubricant next year.

For this reason it is advisable to give consideration from time to time to new lubricants offered by suppliers. Such consideration may take several forms.

First be sure that the supplier definitely recommends the lubricant for the type of application in which you intend to use it. Such recommendation by a reputable supplier leaves little chance that the lubricant will not give at least reasonably good performance.

40" BLOOMING MILL LUBRICATION CHART

Date: 1-1-54

Front Table

PART	METHOD	LUBRICANT	FREQ.
Electric Motor Bearings	Hand Gun	High Stability Grease 1	3 months
Flexible Couplings	Fill Reservoir	E P Grease 0	2 months
Reduction & Line-shaft Gears	Reservoir	E P Oil 110	Daily
Reduction, Line-shaft, & Table Roller Bearings	Centralized System	High Temp. E P Grease 1	Hourly

William Smith
Lubrication Engineer

John Jones
General Foreman Maint.
40" Blooming Mill

Fig. 5 Typical Lubrication Chart.

As a next step, the specifications of the lubricant may be compared with those of the lubricant you are now using and other available similar lubricants. Specifications furnished by the supplier generally are limited to some of the basic physical properties of the lubricant and are sufficient to indicate its type and something of its composition. If a lubricant testing laboratory is available to the plant, further tests which give some indication of a lubricant's performance may be made. These include tests such as those for film strength, grease stability, grease pumpability and the like.

The specifications and test data serve as good means for screening lubricants, but they do not give an absolute measure of the all-around performance of a lubricant in a given application. This can be determined only by observing its performance in actual service as closely as is practical. This is known as the field trial and it should be entered into and carried out jointly by the Lubrication

Engineer and the Maintenance Foreman in the operating unit. In planning a trial, such factors as the possible advantages of the new lubricant, the equipment to be used and the measurements and observations that are to be made should be considered. After completion of the trial, a complete report of the results should be made.

Price, of course, is also a factor in the purchase of lubricants. It must be certain, however, that satisfactory performance has been established before price becomes the deciding factor in procurement. A sacrifice in performance can often make a seemingly low priced lubricant actually become a very expensive commodity. Also, purchase of the highest priced lubricant gives no assurance of superior performance.

If a lubricant has proved itself to be more desirable from an all around standpoint than the one previously in service, a sound basis has been established for changing to it as the regular lubricant.

IMPROVE LUBRICANT APPLICATION METHODS. A highly rewarding phase of any lubrication program in most plants today is the adoption of improved methods for applying lubricants.

Most of the present steel plants went into production near the turn of the century. From time to time over the years these plants have been expanded to increase their capacity and produce new types of products by the addition of new operating units. In many cases the older operating units were retained and still are being used. As a result, a typical steel plant has some modern equipment and also some which is 25 to 50 years old. A similar situation no doubt exists in many older industrial plants.

Usually the older equipment was designed inadequately from a lubrication standpoint. It is characterized by open gearing, plain bearings, lack of reservoirs for lubricant and absence of sealing devices for retention of the lubricant in the bearings. Bearings were lubricated by hand oil can, the oil generally being poured into a hole in the top of the bearing. Most of it quickly escaped and ran onto the floor leaving the bearing virtually unlubricated until the next time the oiler made his rounds. Under these conditions, a high rate of wear takes place both in the bearings and the journals of the shafting. Bearing wear permits displacement of the shaft and improper gear mesh with a consequent high rate of gear tooth wear. Gear teeth were lubricated by periodic hand application of hot gearshield, but this did not provide the continuous lubricant film on the teeth that is required for long gear life.

The older design of equipment in a plant, therefore, represents a fertile field for a program of improving lubricant application by equipping it with modern lubricating devices.

Many centralized grease systems have been installed on the older equipment in steel plants and in almost every case outstanding improvement in lubrication has resulted with the attendant increased life of bearings and gears and with appreciable overall savings. In initiating such a program in a plant, the first few installations showing bearing replacement and maintenance costs before and after installation of the centralized system will convince management of the value of continuing with this type of improvement.

The following types of equipment in steel plants should be provided with centralized grease systems, regardless of their age: (1) All blast furnace tops. (2) All roll neck bearings, plain and antifriction. (3) All front and back mill tables. (4) All reversing tables such as on gag presses and other miscellaneous equipment. (5) All the more important overhead cranes. (6) All other equipment which has frequent repetitive maintenance because of rapid bearing wear.

An essential factor in a program of applying centralized grease systems is the establishment of a small, well-trained group of men in the plant to carry out this work. These same men also may have the responsibility of inspecting and maintaining the grease systems in proper operating condition. The Lubrication Engineer may have the responsibility of collaborating with the Operating Unit Supervisor in planning the installations, seeing that the equipment is ordered and assisting in its installation.

IMPROVE EFFICIENCY OF LUBRICANT HANDLING & STORAGE. The large quantities of lubricants used in the operation of steel plants make the buying and handling of these materials within the plants an important item from the standpoint of housekeeping, safety and costs.

There are several ways in which lubricants are received by the plants and usually there is an appreciable difference in the price of the lubricant depending upon which method is used. Fluid lubricants may be bought in less than carload lots of drums (LCL), carload lots (80 drums), tank wagon (1500 to 3000 gallons) and tank car (6000 to 10,000 gallons). Greases necessarily must be bought in drums and may be in either L.C.L. or carload lot quantities. There is often a price differential of up to $.10 per gallon between the drum price and the tank car price of fluid lubricants. The differential between drum and tank wagon price is usually $.05 to $.07 per gallon. A savings of approximately one-half cent per pound of grease usually can be realized by purchase in carload lots rather than in a smaller number of drums.

To obtain the lowest prices, plants should purchase their lubricants in the largest quantities consistent with the rate of usage and the capacity of their storage facilities. In the case of fluid lubricants purchased in quantities greater than 20,000 gallons per year, tank wagon or tank car delivery generally is preferred. If adequate storage tanks are already available, quantities of as low as 6000 to 10,000 gallons per year can be advantageously bought in tank cars.

In addition to the price advantage, purchase in bulk eliminates the storage and handling of large numbers of drums within the plant, particularly in those cases where the oil is delivered directly to a

tank at the point of consumption in the plant. In cases where it is impractical to provide storage tanks at the points of consumption, bulk storage facilities located at a central oil storage building will permit purchase in tank car quantities and distribution is made by filling drums and transporting them to the point of consumption.

The preferred method for handling lubricants purchased in drums is to receive the drums at a central storage and place them in suitable racks located indoors if possible. Some type of lift truck is necessary for placing the drums on the racks and for their later removal. For plants purchasing sufficiently large quantities of drum lubricants, adequate storage racks are advantageous not only from the standpoint of permitting purchase in car lot quantities but also to aid greatly in promoting good housekeeping in the lubricant storage area.

Many plants now use trucks equipped with hydraulic lifting end gates for transporting drums within the plant. This equipment has improved greatly the safety of the drum handling by eliminating the necessity of manually loading drums on a truck with the help of an inclined skid. The motor operated lift truck also can be used for greater safety in loading drums on a truck.

It is often advantageous to use a tank truck or other type of portable tank for hauling oil from central storage tanks to consuming units, particularly where these units have storage tanks which are part of circulating oil systems, thereby reducing drum handling. In cases where a truck cannot reach the tank at the consuming unit, unloading of the oil may be accomplished by extending a 2″ or 3″ pipe line from the tank to a location accessible to the truck.

An important part of proper lubricant handling is provision for storage of drums at the consuming units while dispensing the lubricants. For this purpose suitable enclosures should be provided for the drums and use made of the various types of excellent dispensing pumps available for removing either fluid lubricants or greases from the drums. Covered containers should be provided for all lubricants during each stage of their handling in order to prevent the entrance of dirt and to maintain a clean and neat appearance.

Good housekeeping can be maintained only if empty drums are returned from the consuming units to the central storage promptly. The drums then should be sorted and placed in an orderly manner to await return to the supplier or to be collected by a drum handling concern.

For identification of lubricants in handling and applying them throughout the plant, the type name should be stenciled on the lid and side of each drum. The lubricant supplier will perform this stenciling according to the instruction on the order from your plant. Generally it is satisfactory for the supplier to retain his own brand name, batch number, company name and other data on the drums. Thus, the same name appears on the drums as on the lubrication charts, and the chance of a mistake in application is small.

ESTABLISH LUBRICANT CONSUMPTION REPORTS. The expenditure for lubricants in a large steel plant may amount to several hundred thousand dollars a year. It is therefore important that some means be established for controlling the use of lubricating materials to be certain that those purchased are used as efficiently as possible and waste or excessive loss of lubricant is held to a minimum.

In lubricating with oil, the problem of waste through leakage is always present. Few pieces of equipment are absolutely oil tight, a small amount of oil normally escaping past shaft seals. However, there are a number of conditions which develop in machinery which cause oil consumption to become excessive. These include the occurrence of cracks in oil reservoirs, failure of oil seals, breakage of pipe lines in circulating systems, excessive loss of oil while draining water from settling tanks and others.

Grease is not lost as easily as oil, and the problem of controlling it usually is somewhat simpler. The chief loss in the case of grease takes place when timers are set for too frequent greasing intervals on automatic centralized grease systems. A return to the proper timer setting will usually eliminate this type of loss.

The main problem is controlling the consumption of fluid lubricants. A basic tool in the conservation of oil is the issuing of a consumption report each month for each operating unit and for all large consumers of oil within the unit. Such a monthly report in conjunction with a budget for oil consumption, based upon operating turns of the equipment, has been found very effective in keeping operating unit personnel conscious of the leakage problem. A higher than normal consumption on the monthly report will generally impel the necessary maintenance action to bring the consumption back into line.

As equipment gets older there is generally an increase in oil loss. However a good preventive maintenance program can do much toward keeping the loss at least within reasonable limits.

SET UP METHODS FOR MAINTENANCE OF OILS IN SERVICE. Oils in service in large circulating systems should be tested periodically to determine their suitability for continued use. Visual inspection of an oil generally reveals little about its actual condition.

A sample of the oil should be withdrawn from the system and several of its characteristics measured in the laboratory. The usual tests conducted to determine the condition of used oils in steel mill equipment include: (1) Viscosity. (2) Neutralization number. (3) Percent water. (4) Percent sediment. (5) Timken value (if an extreme pressure type oil).

The Lubrication Engineer should review the results of the tests and compare them with the suppliers specifications for new oil. Depending upon the nature or degree of departure from new oil characteristics, the Lubrication Engineer either

will approve the oil for continued use or advise the Operating Unit of any action that should be taken in the way of reconditioning or replacing of the oil.

COLLABORATE ON NEW EQUIPMENT DESIGN. An important factor in improving plant lubrication is close cooperation between the Lubrication Engineer and the Engineering Department in the design and specification of new equipment.

From constantly working on lubrication problems on the various plant equipment, the Lubrication Engineer is very familiar with any design shortcomings, from a lubrication standpoint, in existing equipment. He also is familiar with changes that have been made in the equipment to improve lubrication. Therefore he is in a position to render valuable service by reviewing all projected new equipment and making recommendations relative to lubrication designs.

INVESTIGATE & ADVISE ON SPECIAL LUBRICATION PROBLEMS. The Lubrication Engineer should be prepared to act as consultant for any one in the plant who has a particular lubrication problem and he welcomes having all such problems referred to him. He will investigate them thoroughly and make recommendations for their solution. By such actions, the Lubrication Engineer not only will contribute to the operation of the plant through helping to solve specific problems but will gain the respect and good will of many persons throughout the plant. This is important since it assists him in obtaining cooperation that is required to carry out a number of the other phases of the overall plant program.

COLLABORATION IN TRAINING OF PLANT LUBRICATION PERSONNEL. One of the most important phases of a successful plant lubrication program is the training of lubrication personnel in the proper techniques and methods of doing their work. Maintenance Supervisors and the men who perform the lubrication will be able to do a better job of lubricating the plant equipment if they have been properly instructed in their work. The instruction should include information on lubrication principles as well as specific training in the techniques required for lubricating the particular equipment for which the men are responsible.

The Lubrication Engineer generally collaborates with the various operating units and the training division in the plant to set up needed training courses, develop suitable training materials, and participate in the presentation of the instruction.

Patent Abstracts

(Prepared from **OFFICIAL GAZETTE** by Ann Burchick. Printed copies of patents are available at 25c each from: Commissioner of Patents, Washington 25, D. C.)

Reaction Products of Oxidized Oil and PCl_3, Patent No. 2,681,890, by D. Frazier, assignor to The Standard Oil Co.

A method of preparing a soluble oil which comprises reacting phosphorus trichloride with a highly refined petroleum oil oxidized to have a saponification number between about 20 and 80 and a neutralization number between about 9 and 25,, and then treating the reaction product with an aqueous solution of a strong inorganic base in an amount sufficient to neutralize the reaction product but not in an excess sufficent to raise the pH of the neutralized product above about 10.

Synthetic Lubricants, Patent No. 2,681,905, by W. E. Garwood, F. M. Seger & A. N. Sachanen, assignors to Socony-Vacuum Oil Co., Inc.

The process for preparing a stable, viscous oil of high viscosity index, which comprises: reacting, in a non-oxidizing atmosphere at a temperature between about 500 F. and about 750 F. for a period of time from about ten hours to about one-half hour, respectively, a polymeric hydrocarbon reaction product with an elemental substance selected from the group consisting of sulfur, selenium and tellurium, the quantity of said elemental substance being from about 0.1 per cent to about 5 per cent, by weight, of total reactants, and said polymeric reaction product: having a boiling range from about 150 F. to about 930 F., at which maximum temperature at least about 80 per cent by weight thereof distills, having an average molecular weight from about 150 to about 600, and obtained by thermally and non-catalytically heating in the absence of said elemental substance, a charge consisting essentially of hydrocarbons containing at least about 80 per cent by weight of a normal, alpha mono-olefin having between six and about twelve carbon atoms per molecule, at a temperature between about 600 F. and about 750 F. for a period of time from about twenty hours to about one hour, respectively.

Lubrication Treatment of Textile Materials, Patent No. 2,682,486, by W. P. Humphreys, Jr., assignor to Celanese Corp. of America.

Process for the lubrication of textile materials in filamentary form, which comprises adding to an aqueous medium for washing said textile materials a mixture of textile lubricant and an emulsifying agent, which mixture forms an emulsion with said aqueous medium, passing said aqueous washing medium containing said emulsion through a mass of filamentary materials thereby the textile lubricant is removed from the aqueous phase and deposited on the textile materials, the proportions of textile lubricant and emulsifying agent in said mixture being such that when 1 part by weight of said mixture is added to 3 parts by weight of water, it forms an emulsion so unstable as to begin to break within 2 to 6 hours.

Rust Preventing Compositions and Process, Patent No. 2,682,489, by G. Hugo von Fuchs.

A rust preventive lubricating composition comprising a major proportion of an organic lubricant having in solution therein a minor proportion of an ester of an alkyl succinic acid.

Cellulose Base Lubricating Grease, Patent No. 2,682,506, by A. F. Sirianni & I. E. Puddington, assignors to National Research Council.

A lubricating grease composition comprising a lubricating oil thickened to a grease consistency with solvent-exchanged cellulose precipitated from viscose and containing 3 to 60% by weight, based on the cellulose, of a water insoluble metal soap as a waterproofing agent, said grease containing 2 to 10% by weight, based on the weight of the total composition, of a waterproofing alkyd resin of the glyceryl partial ester dibasic acid type.

Silicon Oils Having Low Viscosity-Temperature Coefficients, Patent No. 2,682,507, by M. C. Agens, assignor to General Electric Co.

A liquid composition of matter having a low viscosity-temperature coefficient consisting essentially of (1) a liquid methyl phenyloplysiloxane con-

(Continued on page 289)

MIXED FRICTION CONDITIONS IN LUBRICATION

by D. D. Fuller*

This is a survey-type paper on the subject of "mixed friction." A review of early work in this field is presented as background to a discussion of a recent valuable paper by Wilhelm Altrogge.

It is generally agreed that journal bearings operate in a completely satisfactory manner, under steady load conditions, if they are properly designed and maintained. A hydrodynamic film is developed by the relative motion of the shaft and the bearing so that the rubbing surfaces may be completely separated. By proper selection of the variables[1], load, speed, clearance, lubricant viscosity, and bearing size, a journal bearing may be expected to provide long, trouble-free service life with a minimum of friction and wear.

Frequently, however, bearings are subjected to extreme conditions of operation such as result from overloads, underspeeds, inadequate viscosity (frequently the result of insufficient cooling), or lack of sufficient lubricant supply. As a result, the hydrodynamic fluid film may be partially or completely ruptured. Some degree of metal-to-metal contact then takes place and friction and wear in these bearings rise sharply. If the journal bearing is operating with high loads, this increase in friction due to partial film break-down may result in severe overheating and catastrophic failure. (An example of a heavily-loaded bearing is an automotive-type bearing.) On the other hand, if the loads and speeds are moderate, the bearing may continue to run fairly well in spite of increased friction and rate of wear resulting from fluid-film breakdown.

Actually many industrial bearings operate under such conditions of partially-ruptured fluid films. These are especially prevalent where the volume of lubricant supplied is not sufficient to permit the formation of a complete fluid film in the bearing. Where the lubricant is introduced by drop-feed oilers, wicks, or felt pads, this is usually the case. The penalty is, of course, high friction with some degree of accelerated wear.

The general picture of bearing performance may be shown by reference to Fig. 1. This is a modification of the familiar diagram showing coefficient of friction plotted against the bearing parameter ZN/P. Z is the viscosity of the lubricant in centipoises; N is the speed in rpm; and P is the average pressure in lb/sq. in. based on projected area. The usual diagram includes just the lower half of Fig. 1. The complete Fig. 1 is based on a suggestion by Larsen & Perry[2]. It is noticed that the minimum value for the coefficient of friction has been selected as approximately 0.001. To the right of this minimum value, operating conditions are such that a complete fluid film is developed in the bearing. This is the region of fluid film lubrication.

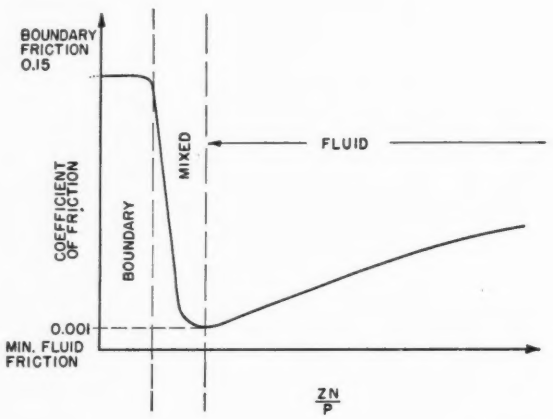

Fig. 1

At the extreme left of Fig. 1 the curve of coefficient of friction levels off at a high value and remains constant. This is recognized as the region of true boundary friction where the coefficient of friction is independent of the viscosity of the lubricant and independent of rubbing speed. Thus for very small values of ZN/P the coefficient of friction remains essentially constant as shown.

Between these two zones, boundary friction on the left and fluid friction on the right, is the zone where with reduction in ZN/P the coefficient of friction rises sharply. There is considerable evidence to indicate that in this zone, a combination of fluid friction and boundary friction prevails. It might be appropriate to call this the zone of mixed friction. Blok[3] refers to this area of mixed friction as the area of "thin film lubrication." The relative contribution of fluid friction as compared to boundary friction would seem to depend upon the point of

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actual operation of the bearing on the friction curve. For example, at the point of minimum friction, the load would still be completely supported by a fluid film. On the other hand, at the top of the steeply rising curve where it has become horizontal the load should, in turn, be completely carried by solid contact through the boundary layers of lubricant. At any point in between these two limits it might be reasonable to conclude that the load would be supported partly by fluid films and partly by direct contact areas of boundary lubrication.

G. B. Karelitz[4] as early as 1926 recognized these conditions of lubrication. His conclusions resulted from a study of the lubrication of waste-packed armature bearings. The tests performed by Karelitz showed that such bearings usually receive an inadequate volume of lubricant supply. As a consequence, the normal hydrodynamic film is ruptured and the contact surface between shaft and bearing contains areas of fluid film pressure and areas of direct contact or boundary friction. The pressure developed in the fluid film areas or fluid film "puddles" is higher than would normally be expected. For example, in a 5" diameter, by 9" long, axle bearing tested by Karelitz at 450 rpm, the applied load was 1200 lbs, and the average pressure based on projected area was about 30 lb/sq. in. However, a fluid-film pressure was measured in the bearing by means of a gauge, 30 deg. behind the load line with respect to the direction of rotation. The actual pressure measured was from 900 to 1000 psi yet the average pressure based on projected area was only 30 lbs/sq. in. The coefficient of friction for the 5" x 9" bearing was measured by Karelitz to be 0.03. He suggested that the percentage of the total load carried by fluid films and the percentage carried by direct contact areas could be estimated by considering the magnitude of the coefficient of friction.

Let P_1 represent that part of the load carried by the oil films and P_2 that part carried by the boundary contact area. Then the total load on the bearing $P = P_1 + P_2$. Karelitz suggested that since high viscosity oil was used in these waste packed bearings, the coefficient of fluid friction could be taken equal to 0.01. The coefficient of friction corresponding to boundary lubrication in the direct contact area could be taken as approximately 0.12. The total friction force on the journal would be the sum of the frictional forces due to hydrodynamic friction and to boundary friction. It is then possible to write the following relationship:

$$P \times f = P_1 \times f_1 + P_2 \times f_2 \qquad (1)$$

where f is the overall coefficient of friction, f_1 is the coefficient of friction for fluid lubrication, and f_2 is the coefficient of friction for boundary lubrication.

Substituting the proposed values of $f_1 = 0.01$, and $f_2 = 0.12$ and the measured value of $f = 0.03$.

$$0.03\,P = 0.01\,P_1 + 0.12\,P_2 \qquad (2)$$

But

$$P = P_1 + P_2 \qquad (3)$$

Substituting (3) in (2)

$$0.03\,[P_1 + P_2] = 0.01\,P_1 + 0.12\,P_2 \qquad (4)$$

or

$$0.09\,P_2 = 0.02\,P_1$$

or

$$P_1 = 4.5\,P_2 \qquad (5)$$

Thus under the above assumptions, for a measured coefficient of friction $f = 0.03$, we would expect that the load carried by the fluid film areas in the bearing would be 4.5 times greater than the load carried on the boundary film areas. To estimate the relative friction contribution of each area to the overall friction of the bearing we can consider that the friction force in the boundary area $= 0.12\,P_2$, the friction force in the fluid areas $= 0.01\,P_1$

But from (5)

$P_1 = 4.5\,P_2$

So the friction force in the fluid areas equals

$0.01 \times 4.5\,P_2 = 0.045\,P_2$

Thus the friction force contributed by the boundary area is $0.12/0.045 = 2.67$ times the friction force of the fluid films. Thus with this approximate analysis we see that, although partial oil films carry most of the load, the direct contact area creates most of the friction.

This condition of partial films in starved bearings has been observed by others including John Boyd[5]. He used a plastic bearing supplied with lubricant from a saturated felt pad. With a colored oil as the lubricant, actual limited areas of fluid films could be seen through the transparent bearing.

Recent research work by Altrogge[6,*] has thrown new light on this aspect of mixed friction. A journal bearing testing machine using a nut-cracker arrangement on two half-bearings was fitted with a piezoelectric pressure pick-up. A pressure chamber was built into the rotating shaft flush with the surface of the shaft. A sensitive transmission chamber converted the pressure from the radial to axial direction and then to a recess in which were placed nine superimposed quartz crystals. The resulting electrical charges were transmitted through a mercury contact and a shielded cable to either a voltmeter or, for high speeds, an oscilloscope.

Fig. 2 shows a photograph of an actual trace for the two half bearings which were first tested under fluid conditions of operation. The ordinates

*Through the interest of Prof. Blok of the Technical University in Delft, Holland, the doctoral thesis of Wilhelm Altrogge was brought to the attention of the Educational Committee of the ASLE. A copy of this thesis was obtained through the kindness of Mr. A. A. Bondi of the Shell Development Co., Emeryville, California. This was translated through the courtesy of the Bell Telephone Laboratories. Copies of the translation are being made available at cost by The Franklin Institute Laboratories for Research & Development, Phila. 3, Pa.

of this trace represent fluid pressures. Experimental conditions are listed in Fig. 2 resulting in a measured coefficient of fluid friction of 0.001.

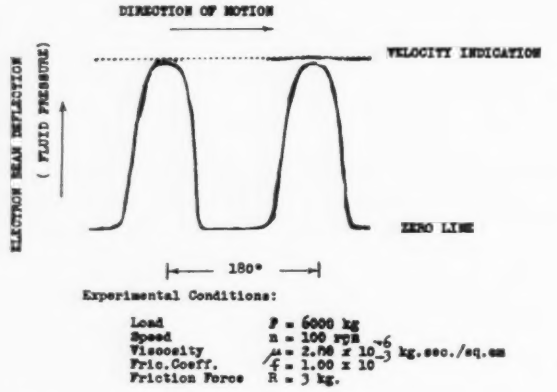

Fig. 2

Fig. 3 shows a corresponding oscilloscope trace for conditions of operation in which the overall coefficient of friction had increased to 0.017. These were presumably conditions of mixed friction, yet the trace indicates that fluid pressures were still present and they behaved in a manner similar to the previous results for complete fluid film conditions. The horizontal line at the top of the figure labelled "standard pressure indication" was obtained in one bearing-half as a check on the piezoelectric pressure indicator. This standard pressure was read on a gauge connected to the bearing-half through a fine capillary tube.

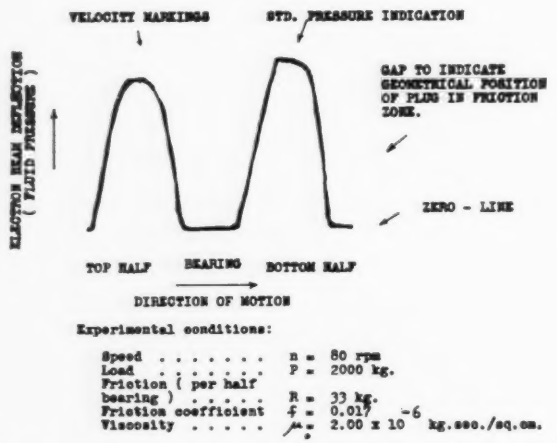
VELOCITY MARKINGS
STD. PRESSURE INDICATION
GAP TO INDICATE GEOMETRICAL POSITION OF PLUG IN FRICTION ZONE.
ELECTRON BEAM DEFLECTION (FLUID PRESSURE)
ZERO - LINE
TOP HALF
BEARING
BOTTOM HALF
DIRECTION OF MOTION
Experimental conditions:
Speed . . . . . n = 80 rpm
Load . . . . . P = 2000 kg.
Friction (per half bearing) . . . . . R = 33 kg.
Friction coefficient f = 0.017
Viscosity . . . . . μ = 2.00 x 10⁻⁶ kg.sec./sq.cm.

Fig. 3

Fig. 4 represents a very interesting set of observations of pressure measurements while the shaft was brought to a gradual stop. At 75 rpm minimum friction was developed. This would correspond to the point of minimum friction in Fig. 1. Then, as the shaft was brought to a stop, the fluid pressure as measured by the piezoelectric pick-up, slowly reduced as shown in Fig. 4. Altrogge concludes that the decrease in fluid pressure represents a gradual transition of the total load-carrying capacity of the bearing from fluid pressure areas to direct contact areas of boundary friction. At zero speed the fluid pressure load-supporting component would be zero — the entire load would then be borne by boundary lubrication areas. Thus, for minimum friction conditions as indicated in Fig. 4, we would conclude that the load is carried completely by a hydrodynamic fluid film. For standstill conditions at zero speed, the load supporting capacity should be predominantly through areas of boundary lubrication.

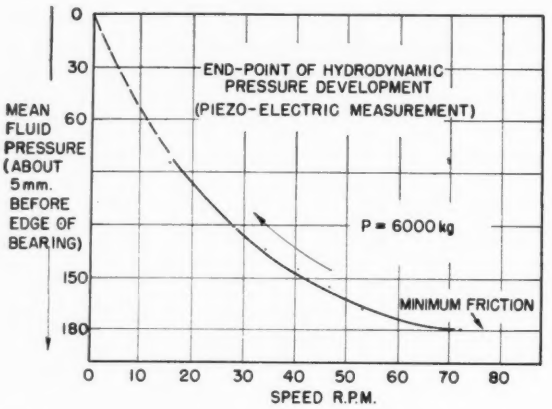

Fig. 4

Fig. 5, also taken from Altrogge's work, is a plot of the observed frictional forces plotted against circumferential speed of the rotating journal. It shows the minimum friction for two loads; one 900 kilograms and the other 500 kilograms. The value of minimum frictional force for both loads is almost identical. However, as might be expected for fluid film conditions, the minima are obtained at different speeds. Then in both cases the shaft is brought to a gradual stop. The instantaneous frictional force was measured by a torque arm fitted with a piezoelectric crystal. As is shown in Fig. 5, while the shaft is brought to a gradual stop, the frictional forces increase.

Approaching zero speed, the frictional force is greater for the higher load and less for the smaller load. The frictional force as such is almost directly proportional to the load. This corresponds to what we would expect under conditions of boundary friction. Thus, the two extremes are shown in Fig. 5. For fluid film conditions the magnitude of minimum friction is independent of the load. For purely boundary lubrication conditions the frictional force is directly proportional to the load. And in between these two limits the frictional force is presumably made up of a fluid friction component and a boundary friction component.

Altrogge developed a very ingenious and apparently accurate method for measuring initial metallic contact in a bearing. He was able to show that for operating conditions to the left of the point of minimum friction, as shown in Fig. 1, partial metallic contact does take place. He developed a

detector to indicate this point of initial solid contact. It is based on the theory of cuprous oxide rectifiers. A cuprous oxide film is placed on the inner bearing surface. A high frequency alternating current is applied to a circuit through the bearing but, due to the rectifying effect of the cuprous oxide film, the electrons pass through this film in only one direction. The flow of current is measured with a DC milliammeter. When metallic contact is achieved, by the reduction in the thickness of the fluid film, initial plastic deformation of the peaks of surface roughness takes place. This breaks down the cuprous-oxide layer. Electrons will then pass in both directions and the pointer of the DC indicating milliammeter drops to zero.

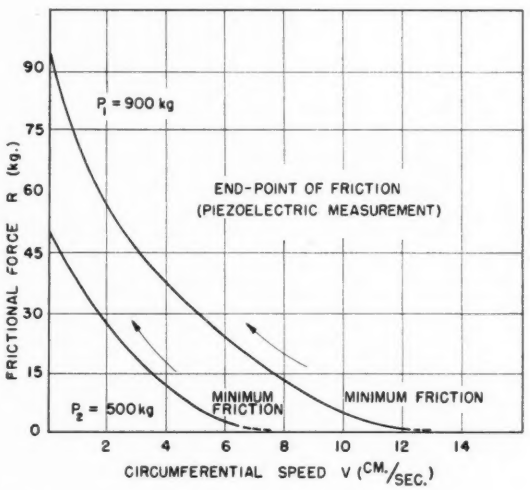

Fig. 5

This apparently is a sensitive and accurate test for marking the point of initial metallic contact. Altrogge found that the maximum DC current flow occurred close to the point of minimum coefficient of friction as represented in Fig. 1. With increased roughness the minimum shifted to the right. With break-in of the rubbing surfaces, the minimum shifted to the left. The minimum coefficient of friction, therefore, is taken to represent the limiting condition of pure fluid friction.

Fig. 6 is a diagram by Altrogge to represent data taken by Stribeck[7] and are actual data similar to the idealized representation of Fig. 1. It is observed that for a given bearing the minimum coefficient of friction for the several loads indicated, is a constant. Stribeck reports however that approaching standstill, all these friction curves converge to a value for the coefficient of boundary friction of 0.14. This would agree in part with the previous analysis and tends to support the data shown in Fig. 5.

It should be pointed out that, in general, conditions of mixed friction are quite common. Even when it is desired to operate with pure boundary friction, great care must be exercised to eliminate possible hydrodynamic action.

Burwell & Strang[8] in their study of boundary lubrication, showed this to be so. They used a test rig comprising crossed cylinders so that the area of contact was exceedingly small and for a given load the resultant pressures were very high.

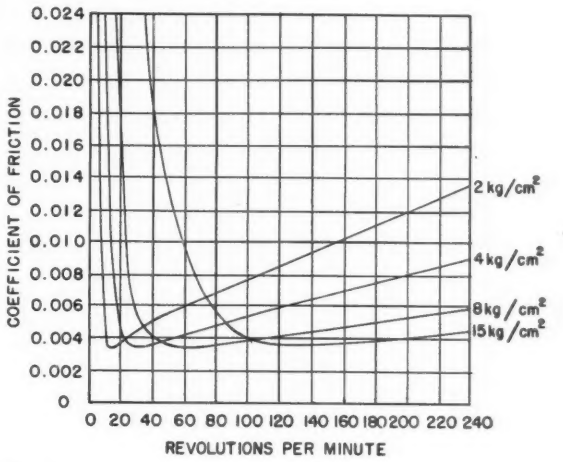

Fig. 6

Fig. 7 shows some of their test results indicating that the coefficient of friction was 0.16 with very low sliding velocities. As the speed was increased the coefficient of friction decreased in the manner shown. They conclude that some hydrodynamic effect must have been present and that it contributed to the load-supporting capacity at the area of contact. Actually, at the test area, conditions of mixed friction probably prevailed. For high sliding velocities, it would be very difficult indeed to eliminate this hydrodynamic component.

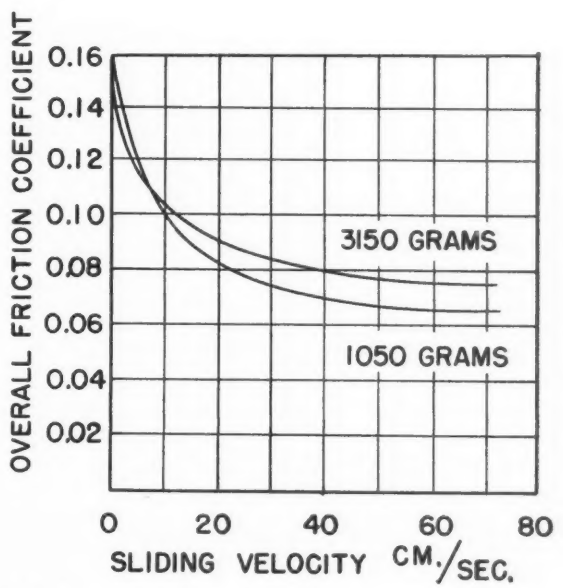

Fig. 7

Many devices have been developed in order to evaluate the effectiveness of boundary lubricants. Unfortunately, hydrodynamic effects may creep in

to partially mask the variables that are being investigated. Machines for testing the extreme-pressure properties of lubricant additives can easily encounter this difficulty. Kolarik, Zeiler & Kipp[9] have discussed this hydrodynamic aspect with regard to the Timken Lubricant Tester. Fig. 8 taken from this reference shows how the Timken "O.K." load is influenced by viscosity.

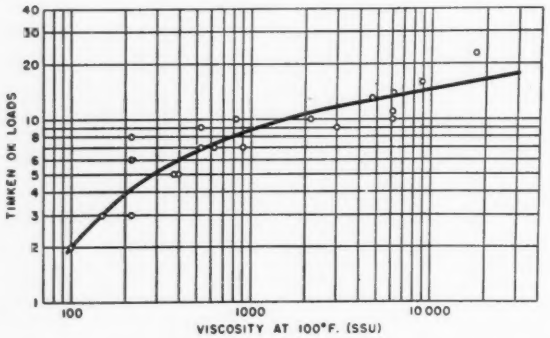

Fig. 8 Effect of variations in viscosity values of straight petroleum mineral oils upon resultant Timken OK Loads.

It appears that the load is carried partially by hydrodynamic action and partially by areas of direct contact under boundary conditions. Even in the absence of oiliness or extreme-pressure agents it was possible to develop the full 100 lbs. maximum "O.K." load value for the Timken machine by manipulation of the variables of viscosity and rubbing speeds. This serves to demonstrate the degree to which hydrodynamic effects may be present with a so-called boundary lubricant tester. When such an apparatus is to be used as a research tool, to achieve basic understanding of the functioning of extreme-pressure lubricant additives, it is apparent that adequate recognition must be given to the possible presence of these hydrodynamic effects.

Larsen [10] discusses the use of the four-ball type testing machine. This is the familiar four-ball machine in which the load is carried on the tiny areas of contact between a nest of three balls at the bottom and one pressing down from the top. Fig. 9 shows typical data obtained by Larsen with the four-ball top. These are highly significant data since they indicate that above a certain critical velocity the coefficient of friction is low, indicating that some kind of hydrodynamic effect is taking place. This occurs in spite of high unit pressures and relatively low sliding speeds. Larsen[2], Beeck[11], and others have recognized these conditions of mixed friction. They called them "quasi-hydrodynamic conditions." They introduced evidence for the existence of this mixed film or "quasi-hydrodynamic" region by a consideration of the coefficient of friction and also by a study of the electrical resistance between the moving parts. Under true boundary conditions electrical resistance was low, of the order of a few ohms. Above a critical velocity, resistance became very high, over 10,000 ohms, even though true hydrodynamic conditions had not yet been reached. It was concluded that the metal surfaces were being supported in some manner by a thin film of lubricant under conditions of load and speed where it is normally considered that a film of lubricant is nonexistent.

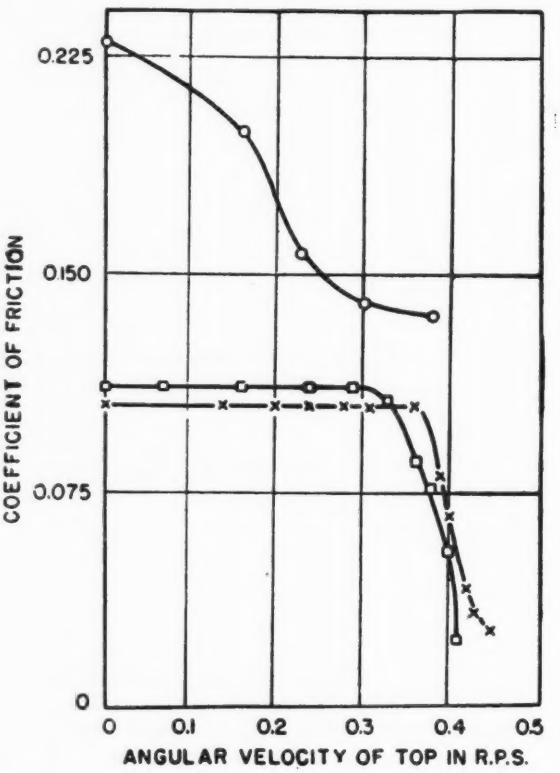

Fig. 9 Friction as a function of velocity in the four-ball top. Key: o = white oil; x = U.S.P. oleic acid; □ = white oil + 1% oleic acid.

CONCLUSION. It becomes evident that there may be a true state of mixed friction in many bearings and other lubricated machine elements. This is the condition where the applied load is supported partially by fluid films and partially by direct contact through areas of boundary lubrication. The relative proportion of the load-carrying capacity contributed by each of these types of lubrication seems to depend upon the severity of operating conditions. True boundary lubrication is considered to exist where the hydrodynamic component is nil. Experimental results may warrant the extension of the hydrodynamic theory into this region of mixed lubrication. Altrogge attempts to do this.

Any investigation of the effect of boundary lubricants should recognize the possibility of the intrusion of natural hydrodynamic action into the performance of test rigs. Recognition of these conditions of mixed friction may well answer some of the questions that have been raised regarding the improvement in performance of journal bearings when oiliness additives are introduced. This may also lead to recommendations for aiding the operation of bearings which are now through necessity

subjected to mixed-film lubrication. The fine research work of Altrogge has contributed much to our understanding of the conditions of mixed-friction.

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Current Literature

(Copies of the following current literature can be obtained without charge by writing direct to the company.)

Series H Pressure Balanced Oil Hydraulic Gear Type Pumps, Bulletin 142. Hydreco Div. (LE 10/5), 1100 E. 222nd St., Cleveland 17, Ohio.

Designed for heavy duty machine tools, materials handling and earth moving equipment, H. Series pumps deliver 40-50-60-70 gpm at 1200 rpm and operate at pressures up to 1500 psi. Chart gives performance details on all four pumps; cutaway drawings show construction details of exclusive Hydreco features.

Preventive Maintenance for Industrial Trucks, by J. A. Draxler. The Elwell-Parker Electric Co. (LE10/5), 4205 St. Clair Ave., Cleveland 3, Ohio.

Details the many requirements of a proper preventive maintenance program for industrial vehicles. Explaining the importance of good lubricants and correct lubrication, and enumerating the principal causes of equipment failure and breakdown, the report also contains several maintenance forms used by Elwell-Parker which may be copied for a company's own use. Additional sections emphasize the vital importance of accurate record-keeping, and development of the proper attitude by employees toward maintenance regulations. A Preventive Maintenance Check List provides a valuable aid to all users of industrial trucks, whether their schedule necessitates weekly, monthly or quarterly check-ups.

Rubber & Leather Packings. E. F. Houghton & Co. (LE10/5), 303 W. Lehigh Ave., Philadelphia 33, Pa.

Six technical bulletins, each featuring essential data on individual types of packings, have been compiled to save time for designers, engineers and maintenance departments. Titles of the various leaflets describe their contents: Rubber Packings, (1) Vix-Syn "O" ring packings and gaskets, plus leather back-up washers, (2) Vix-Syn "V's," (3) Vix-Syn "U" cups; Leather Packings, (1) VIM leather "V's," (2) VIM leather "U's," (3) VIM leather cups.

Liquids Handling Equipment, Bulletin 63. George D. Ellis & Sons, Inc. (LE10/5), American & Luzerne Sts., Philadelphia 40, Pa. (4 pages)

Describes the many features, advantages and specifications of the company's complete line of liquids handling equipment for the petroleum, chemical, mine, railroad and construction industries, including: the side pour type and oval type delivery cans, 100% electrically welded measuring buckets, ten different types of galvanized steel heavy duty funnels of varying sizes, shapes and capacities, and such special equipment as the seraphin test measure, heavy terne plate measures, snap-on-can and faucet tags, "safety" tank truck filling stems, sampling cans, deep seamless boxes for asphalt tests, and brush top cans.

'dag' Dispersions for Industry, Bulletin 460. Acheson Colloids Co. (LE10/5), Port Huron, Mich.

How 'dag' colloidal dispersions serve industry in a wide range of uses—from automobiles to jet engines, and electric toasters to giant machine tools—is thoroughly treated in this new catalog. A centerfold table lists 40 basic dispersions of graphite, molybdenum disulfide, vermiculite, and zinc oxide, in carriers such as water, oil, volatile hydrocrabons, resin solutions, alcohol, glycol, and wax emulsions. Comparative particle size and consistency of each dispersion are given, as well as dilution information and typical applications. 108 specific products and processes aided by 'dag' dispersions are mentioned.

SH & DH High Pressure Lubrication Filters, Bulletin 545. Trabon Engineering Corp. (LE10/5), 1814 E. 40th St., Cleveland 3, Ohio (2 pages)

Includes dimensional material, illustrations, diagrams, and comprehensive data charts on these

(Continued on page 288)

MEASURED OIL FILM PRESSURE DISTRIBUTION IN MISALINED PLAIN BEARINGS

by F. W. Ocvirk*

In the past, many theories have been advanced concerning behavior and oil film pressure distribution of plain bearings with central load. However, much is yet to be determined concerning the behavior of plain bearings with misalinement. Experimental data showing pressure distribution in the oil film of misalined bearings are presented. Both axial and twisting misalinement of a loaded bearing have been investigated. The test results are shown by means of three dimensional plaster models. Since a bearing is rarely installed under conditions of perfect alinement, the following article should be of extreme interest to those concerned with bearing design and operation.

Little quantitative information is available regarding the misalining capacity of lubricated plain bearings. Bearing designers have known that the resistance to journal-bearing tilt is probably small and have therefore attempted to eliminate misalinement where possible. The bearing for an idler with spur gear teeth and offset hub is a case in which misalinement can be eliminated by making the hub central with the gear face. However, if the gear teeth of the idler are made helical, misalinement is introduced because of the components of tooth force in the axial or thrust direction; the axial tooth forces form a twisting couple. In gear applications misalinement is associated quantitatively with moment since tooth forces can be determined. Misalinement in end bearings supporting shafts is related quantitatively to an angular displacement caused by shaft deflection under load.

In order to initiate research on the misalinement characteristics of plain bearings, an investigation was set up at Cornell University under the sponsorship of the National Advisory Committee for Aeronautics. As a first step, it was decided to investigate the hydrodynamic behavior of misalined bearings by measuring pressures in the bearing oil film to determine the nature of the pressure distribution. The results of this experimental program are shown pictorially in the photographs of five plaster models (Figs. 1 to 4) showing oil film pressure distribution. Details concerning the laboratory equipment, the techniques employed, and the methods of reducing data appear in the final report of the investigation: "Experimental Investigation of Oil Film Pressure Distribution for Misalined Plain Bearings," by G. B. DuBois, H. H. Mabie & F. W. Ocvirk, National Advisory Committee for Aeronautics, Technical Note 2507, October 1951.

*Cornell University, Machine Design Dept., Ithaca, N. Y.

This paper was sponsored by the ASLE Technical Committee on Bearings & Bearing Lubrication, and ASME Lubrication Activity, and presented at the ASLE 9th Annual Meeting, Cincinnati, April 6, 1954.

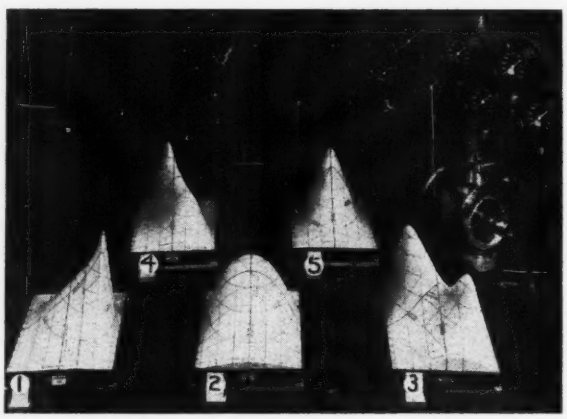


Fig. 1. Plaster models of measured oil film pressure distribution in a 1⅝ by 1⅝ inch plain bearing with a central load of 2,200 lb and various misalining couples. Shown also is the test bearing with pressure gages connected to the oil film.

AXIAL & TWISTING MISALINEMENT. The plaster models show the effect on film pressures of applying misalining moments to a bearing which is also acted upon by a direct central load. To simulate conditions as in the spur gear with offset hub, the misalining moment was applied in the plane of the central load resulting in the pressure distribution of Model 1 (Figs. 1 & 2). This case is considered a case of "axial misalinement." By applying the moment in a plane perpendicular to the central load, as in the case of a helical idler gear, "twisting misalinement" results as shown in Models 3 & 4 (Figs. 1 & 4).

In the models of Figs. 1 to 4, the bearing surface is shown as a developed flat plane in which the shorter dimension represents the width of the bearing and the longer dimension represents 180 degrees of the circumference. Contour lines at 500 psi intervals are drawn on the models to indicate pressure

magnitudes. Because of the large central load of 2200 lb in all experiments, the peak oil film pressures are in the region of 2000-3000 psi. The heavy lines on the models locate the central plane of the bearing and show the circumferential position of the direct central load.

PERCENT MISALINEMENT. In axial misalinement cases, if the misalining moment M is represented by a couple having forces P equal to the central load and a moment arm a, then $M = Pa$. The resultant of the couple and the central load force is a force P offset the distance a from the centerline of the bearing. The ratio of the offset to the width w of bearing is a/w which, if multiplied by 100, may be considered as "per cent misalinement" and is a measure of the magnitude of misalinement.

$$\% \text{ misalinement} = \frac{a}{w} \times 100 = \frac{Pa}{Pw} \times 100 = \frac{M}{Pw} \times 100$$

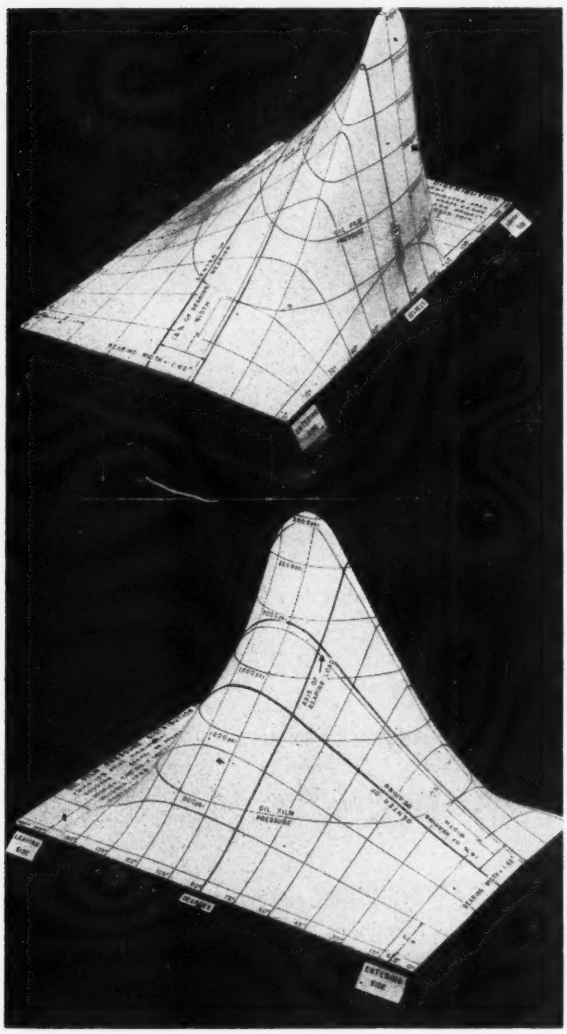


Fig. 2. Model 1, showing film pressure distribution for a plain bearing with axial misalinement represented by a load of 2,200 lb displaced from center 16 per cent of bearing width. Misalining moment, 560 in.-lb; shaft speed, 5,000 rpm; diametral clearance, 0.0033 inch.

For Model 1 the per cent misalinement is 16% representing an offset of the direct load of 1/6th of the bearing width from the bearing center or the placement of the direct load at the third point of the bearing width. Although for cases of twisting misalinement the couple and direct load force cannot be reduced to a resultant force, per cent misalinement may be used as a non-dimensional number (M/Pw) x 100 to indicate magnitude or extent of misalinement. Models 3 & 4 show the effect on pressure distribution for twisting misalinements of 17% and 8%. Models 2 & 5 represent the pressure distribution for direct load without misalinement.

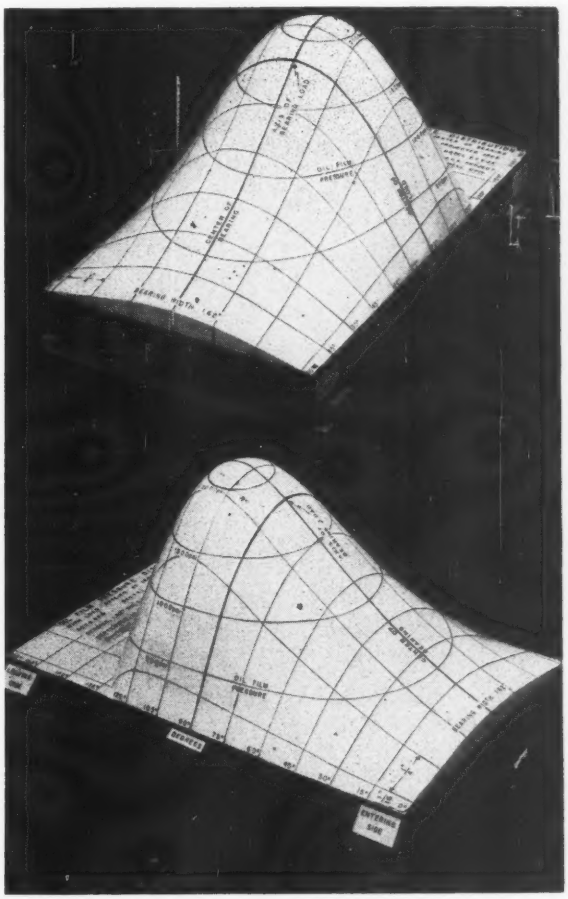


Fig. 3. Model 2, showing film pressure distribution for a plain bearing with a central load of 2,200 lb without misalinement. Unit load on projected area, 833 psi; shaft speed, 5,000 rpm; diametral clearance, 0.0033 inch.

For all models the direct central load was 2200 lb and the speed was 5000 rpm. A comparison of Models 1 & 2 (Figs. 1 to 3) shows that 16 per cent axial misalinement, or an offset of 1/6 the bearing width, raised the peak pressure from 2100 psi to 3100 psi and displaced the peak pressure from the central plane of the bearing to nearly the edge of the bearing. A high pressure near the edge of the bearing may be critical and cause failure of the bearing material such as babbitt. It may be visualized

that the load capacity of a misalined bearing is dependent upon the direct load-carrying capacity of a much shorter bearing near one end.

As would be expected, twisting misalinement results in two pressure peaks, near two locations of minimum film thickness, as shown by Models 3 & 4 (Figs. 1 & 4). Comparing Models 1 & 3 (Fig. 1) having approximately the same percent misalinement, it would appear that a bearing offers somewhat greater resistance to twisting misalinement than to axial misalinement. For a bearing having central load, the twisting couple changes the pressure distribution such as to increase the extent of the working film area whereas the axial couple causes the pressure distribution to shift without an increase in pressurized bearing area.

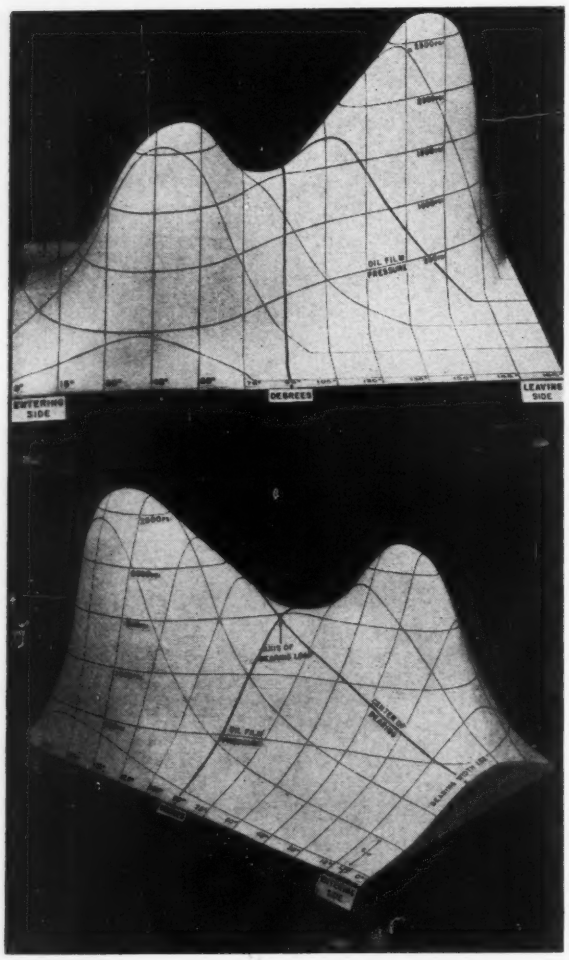


Fig. 4. Model 3, showing film pressure distribution for a plain bearing with a central load of 2,200 lb and 17 per cent twisting misalinement. Misalining moment, 615 in.-lb; shaft speed, 5,000 rpm; diametral clearance, 0.0042 inch.

EFFECT OF BELL-MOUTHING. Considerable bell-mouthing resulted from the misalinement tests in which the high edge film pressures deformed the babbitt. Model 5 (Fig. 1) without misalinement, shows the effect of bell-mouthing on the pressure distribution for central loading. Comparing Models 2 & 5, it may be seen that the increased clearance at the bell-mouthed edges decreased the pressures near the bearing ends and increased the pressures in the central region.

In order to evaluate quantitatively the accuracy of the distributions given by the models, numerical integrations were made of the pressures and moments of pressures for comparison with the loads and couples which were applied in the tests. The most reasonable agreement of integrated and applied values of load is for the case of Model 2 which is without misalinement where the values agreed within 2%. For Models 5, also without misalinement, the agreement is quite close. However, for the cases representing conditions of misalinement, the agreement is less satisfactory. Apparently a closer spacing of the traverse of sampling locations is needed to integrate the sharp peaks accurately.

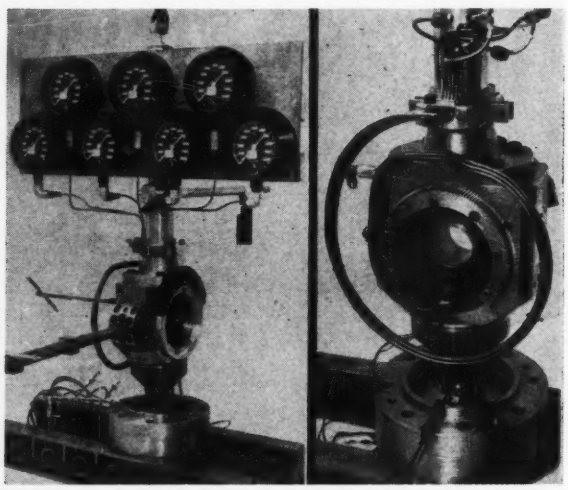


Fig. 5. Test bearing with apparatus to measure oil film pressures. Seven pressure sampling holes in the bearing surface are connected with copper tubes to high pressure gages. Shown also is the gear segment for turning the bearing hub in the bearing housing with a geared key. Central loads are transmitted from a presseure capsule at the bottom through a piston having a hydrostatically pressurized spherical seat in contact with the bearing housing.

EXPERIMENTAL APPARATUS. Superfinished steel shafts and bronze bearings with SAE 13 babbitt lining were used in the experiments. The bearing dimensions were 1⅝ x 1⅝ inches. For Models 1 & 2, the average diametral clearance was 0.0033 inches, and the bearing width was set by shouldering the journal. The bearing rather than the journal was shouldered in the cases of Models 3, 4 & 5 and the diametral clearance was 0.0042 inches. 1120 aviation oil was used as the test lubricant and was fed to the test bearing through two ⅛ inch diameter oil holes 45° apart on the unloaded side of the bearing. The oil was preheated to 140 F., and fed to the bearing at a pressure of 40 psi.

As shown in Fig. 5 the oil film pressures were measured by seven 0.031 inch pressure sampling

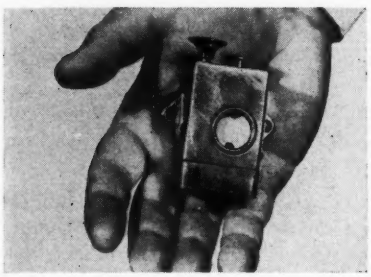
holes located in a line ¼ inches apart along the width of the bearing. High capacity pressure gages, 0-10,000 psi, were connected to the sampling holes. The bearing shell containing the sampling holes could be rotated 65° in either direction from the load line to sweep the bearing surface on the loaded side of the bearing. The sampling holes could be displaced circumferentially while the shaft was rotating by unclamping the bearing block and rotating the bearing shell with a geared key as shown in Fig. 5.

The direct central load was applied hydraulically by a pressure capsule and was transmitted to the bearing through a piston and an oil-pressurized spherical seat. The oil flow through the spherical seat hydrostatically floated the bearing under load giving it freedom to displace angularly in any direction on the application of the misalining couples.

Bearing temperatures were measured by seven iron-constantan thermocouples inserted in 0.094 inch diameter holes to within 1/16 inch of the bearing surface. Spaced ¼ inch apart along the width of the bearing, the thermocouples were located 15° from the pressure sampling holes.

The data points shown in the plaster models are averages of pressures for two directions of shaft rotation and for right and left misalinements. For complete numerical analysis of the bearing systems herein described, the reader is referred to the complete NACA report listed above.

New Products



MINIATURE LUBRICATION SYSTEM, "L Special." A tiny, compact lubricating system especially designed for small machine elements has been developed that brings the advantages of low-cost automatic lubrication to machines having minimum lubrication demands. Comprising all the elements of a complete lubrication system — reservoir, filter, pump, and meter-units — it measures but 2¼" high x 1⅜" deep x 1½" wide. The reservoir holds 20 cc and the pump can be adjusted to discharge a total of .6 cc, .45 cc, or .3 cc each time the plunger is pulled. The "L Special" may be installed to discharge into passages drilled in the machine and leading to the lubrication points, or it may discharge into tubing leading to these points. Of polished diecast aluminum, a one-piece ⅝" diameter window in front of the reservoir acts as a sight gauge to indicate oil level within the reservoir. For additional information, write: Bijur Lubricating Co. (LE10/5), Rochelle Park, N. J.

VISCOSITY MEASURING INSTRUMENT. The Cincinnati Division of Bendix Aviation Corp., has made some important advancements in its new viscosity measuring instrument, the "Bendix Ultra-Viscoson." Utilizing ultrasonic principles, it is now offered in a complete line of indicating, recording and control systems which feature a new approach to process control through viscosity. Employed successfully to control fuel oil blending, fractionation of lubricating oils, asphalt oxidation and asphalt cutback in petroleum refineries and fuel atomization in conjunction with open hearth furnaces in the steel industry, marked success has also been achieved with laminating resins, gravure inks and interface detection. For complete details, write: Bendix Aviation Corp. (LE10/5), 203 W. 3rd, Cincinnati 2, Ohio.

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LOW TORQUE LEATHER OIL SEAL. A new leather oil seal, "National Micro-Torc," has been developed that provides extremely low torque, cool operation, no measurable leakage and exceptional life even at 200 F. Con-

(Continued on page 282)

SYNTHETIC AIRCRAFT GREASES

by H. Schwenker, J. A. King & J. C. Mosteller*

Arylurea-silicone greases have shown the most promise for use over the —65 to 450 F. temperature range. While good results have been obtained with these greases in the ABEC-NLGI bearing tester at 450 F., they do not meet the gear wear or rheological requirements at —65 F. Development of a 700 F. grease has been primarily devoted to the evaluation of high temperature thickeners. Bentone 34 and lithium 9/10 hydroxystearate have been investigated as thickening agents in synthetic type oils. Greases prepared from these thickeners and oils have been evaluated and their applicability with respect to use in aircraft greases determined.

Before the jet age, temperatures encountered in aircraft service were dictated largely by the natural environment and generally ranged from —65 to 200 F. Occasionally in special applications a temperature as low as —100 F. might be reached. Petroleum base products had all of the requirements necessary to lubricate the equipment operating within these temperatures.

The advent of the jet era was accompanied by a drastic increase in the high temperature limit. Increased speed, miniaturization of the equipment and larger power plants have caused the maximum operating temperatures to be extended to 450 F. and even higher. Consequently, aircraft greases now must perform satisfactorily throughout the temperature range extending from —65 F. (and sometimes —100 F.) to 450 F. and above.

The ideal solution to the current greases lubrication problem would be the development of a single grease capable of providing adequate lubrication over this whole temperature range. The development of such a product has been stymied because the mineral oils, which normally are used in the manufacture of greases and which have the desirable low temperature characteristics, do not possess the required thermal stability at temperatures as high as 450 F.

ARYLUREA-SILICONE OIL GREASES. The Air Force contracted with the Standard Oil Company (Indiana) for the development of a grease which would operate satisfactorily over the temperature range from —65 F. to 450 F. The proposed specifications for this product are presented in Table I.

A silicone oil was selected as one having the best high temperature characteristics, and greases were prepared with this material using both organic and inorganic type thickeners. The high temperature performance characteristics of these experimental products were determined by the ABEC-NLGI bearing tester operating at 450 F. and 10,000 rpm. The test results indicated that the organic type thickeners were the most satisfactory and of those, the aryl-substituted ureas were the best.

Actually, arylurea-silicone oil greases have been developed which meet all but three of the proposed requirements given in Table I. These include the low temperature torque, the gear wear and apparent viscosity. Again the limitations of the base oil are responsible for the failure to meet these specifications.

Low temperature operation of these greases would be limited to —40 F., for most applications, although satisfactory performance could be extended to —50 F. if a reasonable amount of starting torque is available. The poor wear properties of the silicone oils is well known and this chracteristic disadvantage was evident in the arylurea greases. In spite of this, however, these greases gave outstanding performance in the ABEC-NLGI bearing tester at 450 F., with several of them consistently exceeding 600 hours.

The most promising arylurea greases have been distributed to the Co-ordinating Research Council for correlation tests on the high temperature bearing tester. Factors such as the type of surfaces involved, high speeds and heavy loads may cause a grease to perform satisfactorily in one 450 F. application but fail in another. Samples of these greases have been distributed also to manufacturers of aircraft equipment for service evaluation tests. It is hoped that these service evaluations will make it possible to determine those purposes for which these greases are most suitable. So far, only a few reports have been received from these tests.

In one instance a comparison was made between an arylurea grease (MLG-9300) and a diester-lithium grease (MIL-G-3278) as lubricants in the motor bearings of two types of electro-mechanical actuators, both of which were subjected to ambient temperatures of 250 F. The arylurea grease proved to be the superior of the two and extended the bearing life from 250 hours to over 400 hours. Consequently the manufacturer has approved the arylurea-type grease as the accepted lubricant for these actuators.

On the basis of the promising results obtained, it was decided to consider the commercial production of these greases. One of the big hurdles to be overcome was that concerning the availability of the raw materials used in the preparation of the arylurea thickeners. The ingredients employed in the preparation of the experimental products were highly purified and were not available in quantity. How-

*Wright Air Development Center, Materials Laboratory, Dayton, Ohio.

This paper was sponsored by the ASLE Technical Committee on Lubrication Fundamentals, and presented at the ASLE 9th Annual Meeting, Cincinnati, April 5, 1954.

ever, through the efforts of the Standard Oil Company (Indiana), an arylurea grease (MLG-9305) was developed which can be made available commercially and which will perform satisfactorily throughout the temperature range from —40 F. to 450 F.

Based on the information to date, it appears that the arylurea-silicone greases can be recommended for ball and roller bearing applications in the temperature ranges where soap type petroleum oil or soap type synthetic base oil greases are not applicable. At the moment these greases cannot be recommended for lubricating either sliding surfaces or ball and roller bearings when the speed factor exceeds 200,000.

Efforts are continuing to improve the properties of these greases, particularly the wear characteristics, by the use of additives. It seems that the amount of wear obtained will vary with the test conditions and with the particular metal combination being lubricated[1].

Samples of improved silicone oils have been submitted to the Materials Laboratory, Wright Air Development Center, for evaluation as base oils for these arylurea-type greases. Preliminary investigation indicates that two of the experimental fluids will perform satisfactorily over the temperature range from —65 F. to 450 F. These fluids now are being employed in the further development of the arylurea-silicone greases.

It is expected that future aircraft greases may be required to withstand temperatures as high as 700 F., and work is being devoted to this problem. Since there are no commercial or experimental oils currently available which are capable of performing satisfactorily at this high temperature, potential thickeners are being evaluated in the present synthetic type lubricants. It is hoped that satisfactory thickeners will be ready for use with the 700 F. fluids when they become available. Considerable effort is being expended by the Materials Laboratory toward the development of these fluids.

BENTONE 34 GREASES. Bentone 34 has been evaluated by the Materials Laboratory as a thickening agent for synthetic aircraft greases[2]. This material is a finely divided, light cream colored powder and is the product of a reaction between an organic ammonium cation and the clay, montmorillonite. Various synthetic oils were employed and the greases were prepared on a small scale, with no batch exceeding four pounds. In preparing the grease, Bentone 34 and the base oil were mixed together at room temperature, after which the resulting mixture was heated at 200 F. for two hours in order to obtain maximum gelation. The finished grease was produced by milling this material on a three-roll mill. Any additives included were added to the oil before it was mixed with the Bentone 34. Approximately 20 weight per cent of this thickener was required to gel most oils to the point where a

Test	Requirements	Test Method
1. Penetration	Worked penetration of 260-310.	Spec. VV-L-791, Method 31.1.4
2. Dropping Point	As determined.	Spec. VV-L-791d, Method 142.1
3. Dirt Count	No abrasive foreign matter. Non-abrasive particles limits; 7500 max. per $cm.^3$ of 25 microns diam. or above. 1600 max./$cm.^3$ of 75 microns diam. or above. None of 125 microns diam. or above.	Spec. VV-L-791d, Method 300.5.1
4. Low Temperature Torque	15 seconds max. at -65 F. under 2000 gm.-cm. load.	See Spec. AN-G-25
5. Apparent Viscosity	15000 poises at -65 F. at a shear rate of 20 reciprocal seconds.	See Spec. AN-G-25
6. Bomb Oxidation	5 PSI pressure drop at 250 F. after 100 hours.	Spec. VV-L-791d, Method 345.3
7. Copper Corrosion	Slight stain allowable. No brown stain or pitting.	Spec. VV-L-791d, Method 530.9
8. Gear Wear	Not more than 2.5 mg. per 1000 cycles under 5 lb. load nor more than 3.5 mg. per 1000 cycles under 10 lb. load.	See Specification MIL-G-3278
9. Evaporation Loss	4% maximum after 22 hours at 400 F.	ASTM Method D-972-48T
10. High Temperature Endurance	Shall satisfactorily lubricate a 204-K bearing at 400 F. for 500 hours. Tests at 450 F. to be performed at the option of the project engineer.	Spec. VV-L-791, Method 33.1
11. Stability After Mechanical Working	Shall not exceed a penetration of 375 after 100,000 worker strokes.	Spec. VV-L-791d, Method 31.3

Table I Requirements for —65 to 450 F. grease.

grease of good consistency was obtained.

The compositions of the greases tested are presented in Table II, together with the results of the tests of their mechanical stability.

The water resistance of the greases was excellent as determined by the Water Resistance and the Water Immersion Tests.

Oil separation seemed to be a function of both the type of oil and the amount of thickener employed. In general, resistance to bleeding was good. Furthermore, evaporation losses of these greases at 210 F. was found to depend entirely upon the base oil used. The Bentone 34 neither accelerated nor retarded the original rate of evaporation of the base oils.

Those greases containing oxidation inhibitors exhibited good oxidation resistance when tested by the Bomb Oxidation Method at 210 F.

Wear characteristics were determined by both the Navy Gear Wear and the Shell 4-Ball Wear Tests. The results shown in Tables III & IV indicate that the greases have good wear properties.

The rust preventive characteristics of the Bentone 34 greases were evaluated in a humidity cabinet under high relative humidity conditions. The uninhibited Bentone 34 greases exhibited very poor rust protection properties and were markedly inferior to uninhibited greases prepared with soap type thickeners in the same base oils. The best result obtained with rust inhibited Bentone 34 greases was a No. 2 rating for the test bearing after 28 days in the humidity cabinet. Attempts continue to be made to improve the rust preventive properties of these greases.

Oil	% Bentone-34	% Additive	Weight Loss of Gears mg/1000 cycles	
			5# Load	10# Load
Orsil BF-1-S Inhibited Silicate Ester	20	?	1.5	3.5
DC-550 Silicone	20	--	1.7	4.8
Di(2-ethylhexyl) sebacate	20	--	1.2	3.7
LB-135-X Inhibited Polyethylene glycol	20	?	0.9	2.0
1010 Base Petroleum Oil	20	--	0.8	1.3
Dipropyleneglycol dipelargonate	20	1% Ortholeum 300	0.6	1.5
Dipropyleneglycol dipelargonate	22	2% Bryton HY Sulfonate	0.7	1.9
15% Tridecyl azelate 85% Dipropyleneglycol dipelargonate	24	1% Dilauryl Selenide 2% Calcium Sulfonate	0.9	3.5

Table III Navy gear wear tests of Bentone-34 greases.

Oil	% Bentone-34	% Additive	Initial Micro-penetration	Micro-penetration after 500 strokes Microworker	ASTM Unworked Penetrat.	Worked Penetration	Worked Stabil. Penetration
Orsil BF-1-S Inhibited Silicate Ester	20	?	102	70	--	--	--
DC-550 Silicone	20	--	65	108	--	--	--
Di(2-ethylhexyl) sebacate	20	--	67	62	--	--	--
LB-135-X Inhibited Polyethyleneglycol	20	?	135	156	--	--	--
Dipropyleneglycol dipelargonate	22	2% Bryton HY Sulfonate	66	83	287	299	315
DC-710 Silicone Oil	12	--	88	181	--	--	--
Dipropyleneglycol dipelargonate	20	1% Ortholeum 300	--	--	279	290	--
15% Tridecyl azelate, 85% Dipropyleneglycol dipelargonate	24	1% Dilauryl selenide Calcium 2% Sulfonate	--	--	224	250	280
85% Dipropyleneglycol dipelargonate 15% Orsil BF-1-S	20	1% Dilauryl selenide 2% Calcium Sulfonate	--	--	290	305	327

Table II Composition and mechanical stability of Bentone-34 greases.

Four High Temperature Performance Tests at 250 F. were made on a Bentone 34 grease which did not contain an oxidation inhibitor. Results of the individual runs varied from 240 hours to 663 hours with the average being 392 hours. Similar tests were conducted on these greases containing an oxidation inhibitor, and the results are shown below:

	Run No. 1	Run No. 2
Bentone 34, Diester-Silicate Ester	810 hrs.	1022 hrs.
Bentone 34, Diester	1005 hrs.	1005 hrs.

The runs made on the diester-silicate ester grease failed at the hours indicated. The bearings were badly coated and would not rotate freely, and the grease was reduced to a gum-like residue. On the other hand, the two runs with the diester grease were arbitrarily discontinued at the end of 1005 hours with the grease and the bearings still in excellent condition.

The Bentone 34 greases containing low temperature base oils were satisfactory at a temperature as low as —65 F. as determined by the torque tester. One Bentone 34-Diester oil grease also passed the Instrument Performance Test at —65 F.

On the basis of the information obtained to date, it appears that Bentone 34 may have application as a thickening agent for synthetic base greases such as MIL-G-3278 as well as high and low temperature aircraft and instrument applications, providing their rust protection characteristics can be improved and providing their performance in high speed bearings prove satisfactory.

LITHIUM 9 or 10 HYDROXYSTEARATE GREASES. Some of the MIL-G-3278 and MIL-G-7421 greases employ lithium 12-hydroxystearate as the thickening agent. Since the 12-hydroxystearate acid is obtained from castor oil which is an imported material, it is always possible that this raw material may become unavailable. On the other hand, the 9 or 10-hydroxystearate acids may be prepared from domestic materials which are readily available.

(Test Conditions: 2 Hours, 1800 RPM, 10 Kg. Load) Composition of Greases	Wear Spot Millimeters
20% Bentone-34, 79% Dipropyleneglycol dipelargonate, 1% Ortholeum 300	0.61
20% Bentone-34, 80% LB-135-X Inhibited Polyalkyleneglycol	0.64
120 Lithium 12-hydroxystearate, 88% Orsil BF-1-S Inhibited Silicate Ester	0.52
16% Silica Gel, 84% Dipropyleneglycol dipelargonate	0.87
MIL-G-3278 Grease (Lithium Stearate, Diester oil)	0.74
MIL-L-7711 Grease (Na, Ca, Petroleum oil)	0.56

Table IV Shell 4-Ball wear tests on soap and non-soap greases.

Oil	% Soap	Initial Micro-penetration	Micropenetration After Working in Hain Microworker		
			50 Strokes	100 Strokes	500 Strokes
Di(2-ethylhexyl) sebacate	12 Soap A	62	74	71	94
Di(2-ethylhexyl sebacate	12 Soap B	164	257	248	260
Di(2-ethylhexyl) sebacate	12 Soap C	81	129	143	207
Dipropyleneglycol dipelargonate	12 Soap A	45	56	59	72
Dipropyleneglycol dipelargonate	12 Soap B	72	100	122	140
Dipropyleneglycol dipelargonate	12 Soap C	65	89	108	130
Tridecyl azelate	12 Soap A	42	52	57	69
Tridecyl azelate	12 Soap B	72	100	122	140
Tridecyl azelate	12 Soap C	66	101	115	142

Table V Composition and mechanical stability of lithium hydroxystearate greases.

The possibility that the lithium soaps of these acids may be employed as substitute thickeners has been investigated. Samples of both the 9/ 10 and the 12-hydroxystearate acids were obtained from Emery Industries and another sample of the 9/10-acid was procured from the Department of Agriculture.

Fig. 1 Unworked lithium 12-hyroxystearate grease. Mag. 32,100X

These acids were converted into three lithium soaps which had the following melting points:

Material	Melting Point	Source of Acid
(a) lithium 12-hydroxystearate	207-208.5 C.	Emery Industries
(b) lithium 9/10-hydroxystearate	201-204.5 C.	Emery Industries
(c) lithium 9/10-hydroxystearate	207-210 C.	NRRL Dept. of Agriculture

These thickeners were used with the following oils to make greases:

Oil	Source
(1) di (2-ethylhexyl) sebacate	Rohm & Haas
(2) tri-decyl azelate	Emery Industries
(3) di-propyleneglycol dipelargonate	Emery Industries

The composition of some of these greases and the mechanical stabilities are presented in Table V. The lithium soaps of the 9/10 hydroxystearate acids yielded a somewhat softer grease than that obtained with the same quantity of the 12-hydroxystearate when used with the same base oil. Although the gelling power of the 9/10-hydroxystearates was less than that of lithium 12-hydroxystearate, they do appear to be good gelling agents.

The Hain Micro worker was employed to determine the mechanical stability of the greases. The 12-hydoxystearate greases exhibited excellent mechanical stability whereas those prepared with the lithium 9/10-hydroxystearate were only fair in this regard. The electron-micrographs of the greases revealed that the 12-hydroxystearate grease (Fig. 1) had the twisted symmetrical type of fiber structure which is characteristic of this type of grease and which has been associated with the good mechanical stability. On the other hand, the 9/10-hydroxystearate greases had rod-like structures and the fibers were packed together in bundles (Fig. 2). It had been supposed that the twisted symmetrical fiber structure, which accounts for the stability of the 12-hydroxystearate greases, was due merely to the presence of the -OH group in the stearic acid molecule. However, it now appears that this structure is due not only to the presence of the -OH group but also to its location.

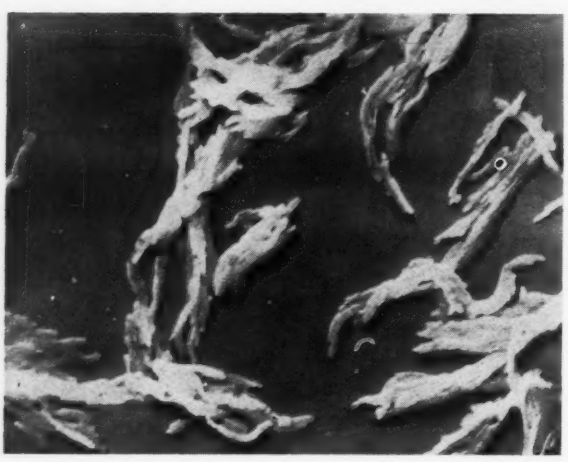

Fig. 2 Unworked lithium 9/10-hydroxystearate grease. Mag. 32,100X.

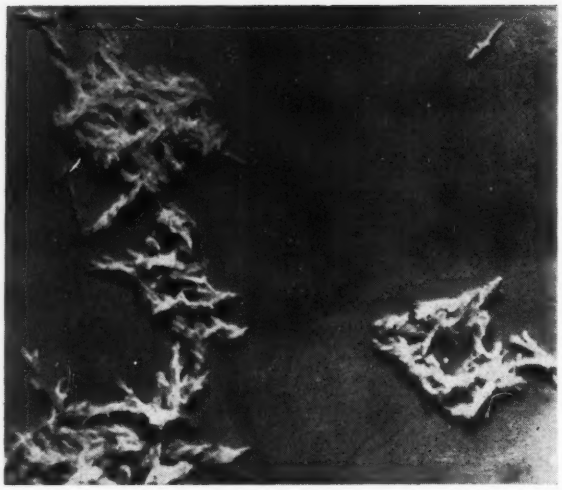

Fig. 3 Worked lithium 12-hydroxystearate grease. Mag. 32,100X.

The fibers of both the 12 and the 9/10-hydroxystearate greases are packed closer together after working. Those of the lithium 12-hydroxystearate grease do not form a solid mass but rather retain a lattice type structure with many interstices which still make available a large amount of the thickener for retention of the oil (Fig. 3). On the other hand, the fibers of the lithium 9/10-hydroxystearate greases form a solid mass upon working and thus reduce the effective amount of thickener available to retain the oil (Fig. 4). This may explain the differences in mechanical stability of the two types of greases.

Oil separation was greater with the 9/10-hydroxystearate greases than with those prepared

Fig. 4 Worked lithium 9/10-hydroxystearate grease. Mag. 32,100X.

88% Oil -- Type Oil	12% Thickener -- Type Soap	Loss Wt. mg/1000 cycles 5# Load	Loss Wt. mg/1000 cycles 10# Load
Di(2-ethylhexyl) sebacate	Lithium 12-hydroxystearate	0.63	1.27
Di(2-ethylhexyl) sebacate	Lithium 9/10-hydroxy-stearate	0.54	1.15
Dipropyleneglycol dipelargonate	Lithium 12-hydroxystearate	0.55	1.22
Dipropyleneglycol dipelargonate	Lithium 9/10-hydroxy-stearate	0.46	1.40

Table VI Navy gear wear tests hydroxystearate greases.

with the same quantity of lithium 12-hydroxystearate in the same base oils. Undoubtedly the amount of oil separation could be reduced by increasing the percentage of lithium 9/10-hydroxystearate so as to yield a grease of the same consistency as that of the 12-hydroxystearate greases.

The water resistance of the 9/10-hydroxystearate greases was considerably less than that of the 12-hydroxystearate greases, as determined by the Water Immersion and the Water Resistance Tests. Even so, the water resistance of the 9/10 hydroxystearate greases was considered adequate for most aircraft applications.

Results obtained on the Navy Gear Wear Tester are shown in Table VI. There was no appreciable difference in the wear characteristics of the greases, and their wear properties were considered to be good.

The melting points of both the 9/10 and the 12-hydroxystearate greases were approximately the same and ranged from 350 F. to 370 F.

The greatest problem to be overcome with regard to the lithium 9/10-hydroxystearate greases is their mediocre mechanical stability. If this can be corrected, another soap type thickener will be available from domestic sources for use in aircraft lubricating greases such as MIL-G-3278. Work is currently underway in an attempt to solve this problem.

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Lubrication in the News

OIL DESIGNED FOR MODERN ENGINES. Development of a new oil designed for modern engines was described by L. Raymond of Socony-Vacuum Oil Co. and J. F. Socolofsky of Socony-Vacuum Laboratories at a recent SAE meeting. The problem was to develop an oil forming less (and different) combustion-chamber deposits in order to reduce octane-requirement increase, pre-ignition, and spark-plug misfiring without sacrificing other objectives of reduced wear and engine cleanliness. Test data indicated that increasing percentages of bright stock in oils led to higher ratios of carbonaceous material to lead salts in the engine deposits. This was believed largely to explain the higer octane requirement observed with such oils. Oils of increased viscosity also raised octane requirement.

Rather limited data indicated no significant effect on octaine requirement from use of detergent additives in oils when the engine was fueled with leaded commercial gasoline. However, with unleaded alkylate, a non-detergent oil gave a much lower octane-requirement increase than did a Series 2 oil. This lead the researchers to believe that generalizations on the effect of additives are probably premature. Individual results may differ, depending on the particular base stocks, additives, engines, fuels, and test conditions used.

The oil finally developed was of the multigraded type, comprising a blend of selected distillate oils plus a combination of viscosity-index improvers and a new detergent-inhibitor combination. Not only did the new oil reduce the octane-requirement increase of engines ititially having clean combustion chambers, but its addition to 'carboned' engines reportedly effected an average reduction in octane requirement of approximately three numbers. As compared with conventional SAE 20W oils in laboratory tests, the oil gave a much lower preignition requirement (expressed as octane number) in one engine, a great reduction in the incidence of preignition in another, and complete elimination of wild ping in a third. In road tests it showed advantages in several forms of uncontrolled combustion, including reduced knock, fewer wild pings, and no after-running.

NEW STANDARDS. Quart motor oil can dimensions have been made uniform in a new American Standard: B64.1-1954. This standard will benefit the packaging industry, oil companies, service stations, and manufacturers of cans and machinery for making and filling cans. Oil companies will no longer have difficulty in running cans through their filling machines because of varying sizes; those who package cans may use standard size cartons; service station workers may neatly and conveniently arrange cans on standard size display racks; can companies need make only one standard dimension of can, and even manufacturers of such machinery need make only one type of machine.

MULTIGRADE OILS POSE OTHER PROBLEMS, as well as offer advantages, C. W. Georgi of Quaker State Oil Refining Corp. pointed out at a recent SAE meeting. Will a 10W-30 oil prove suitable for use where an SAE 20 oil is needed, since the multiple grade may meet the viscosity ranges for 10W, 20W and 30, but not for SAE 20? Other multigrade oils, such as 20W-30 and 20W-40, also involve skipped grades. Actually, a 5W-50 oil could be made from a low-viscosity base stock by heavy addition of viscosity-index improver. However, the resulting oil would have extremely poor consumption characteristics and might fail to fulfill its primary function of lubrication.

ENGINE BEARING DESIGN TODAY

by J. B. Bidwell*

Engine bearings are only one facet of the complete internal combustion engine design problem. The goal of the engine designer is to devise a compact, durable unit which will economically produce the desired power output. The configuration of the bearings is perhaps the most compromised and restricted element in the whole engine—the sizes are dictated by the space available rather than the size desired by the bearing engineer. This paper is not intended to be a complete guide to the design of engine bearings, but rather to point out a number of problems peculiar to engine bearings as a consequence of the operating conditions encountered.

ENGINE DESIGN PROCEDURE. Since the purpose of an engine is to produce useful work by conversion from heat obtained from burning fuel, first consideration is given to the design of the combustion chamber. Selection of the number of cylinders and the arrangement of cylinders of the required size to provide the desired power output is based upon the ultimate use of the engine and the space available. The ratio of cylinder bore to stroke which must be established at this time has a profound influence upon the bearing design, since it bears heavily upon the loads encountered and the space available. The cylinder spacing necessary to provide room for the bore, cylinder walls, and water passages may then be approximated. This same space must then be allotted to the main bearings, rod bearings, crank cheeks, and fillets. Compromises must be made between thick cheeks and generous fillets to produce a strong, rigid shaft and adequate bearing area to avoid bearing problems.

BEARING LOADING. Loads applied to the bearings as a result of the combustion process are used to establish the shaft sizes needed to support the torsional and bending loads encountered. With the tentative bearing sizes established, the next step is to determine as exactly as possible not only the maximum loads, but the complete load diagrams. If pressure cards are available from single cylinder tests of the combustion chamber, the rod bearing loads may be found quite accurately. The load diagram shown in Fig. 1a takes into account not only the gas loads, applied to the piston head, but also the inertia loads resulting from the reciprocation of the piston assembly and rotation of the crankshaft and rod. All of these loads are affected by engine speed so that several conditions must be considered. The maximum load applied to the rod bearing at 4000 RPM, as shown, is about 3300 pounds and occurs at 0° which is top center on the exhaust stroke. However, at 1800 RPM the inertia loads are reduced and the maximum load is about 5800 pounds. At this speed it will occur at about 375° or just after top center on the expansion stroke. This load will be directed upward on the bearing. The values quoted here are for a typical V-8 engine of 7-1 compression ratio with a bore of 3¾ inches and a stroke of 3½ inches.

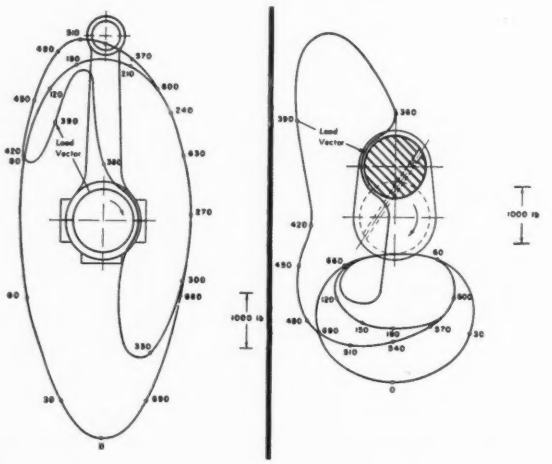

Fig. 1a (left) Rod Bearing Load—4000 rpm; Fig. 1b (right) Crank Pin Load—4000 rpm.

Once the rod bearing loads are determined, the main bearing loads may be approximated if certain assumptions are made. The crankshaft is considered to support no moment in the main bearing journal. This approximation is necessary, since deflections of the crankcase and shaft are indeterminate. Main bearing loads are then the resultant of loads from the two immediately adjacent spans. In addition to the loads applied by the connecting rod, the force produced by rotating mass of counterweight applied to the shaft must be included. Such a load diagram is shown in Fig. 2a. The crank angles here are measured from top center on the exhaust stroke of cylinder No. 1. Like the rod bearing loads, the load applied to the main is also dependent upon speed. At 1800 RPM the maximum load occurs at 555° crank angle and is about 4300 pounds. The highest loads are applied to the lower half of this main bearing throughout the speed range of the engine considered here.

The space allotted to the bearings may be now reassessed in the light of the calculated bearing loads. Only if no compromise can be made between crank cheek and bearing width can considera-

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tion be given to increasing bore spacing beyond that required for the bores and water passages, since over-all size is increased by such a change. The minimum crank cheek thickness is dependent upon the shaft stiffness required to prevent "roughness" of the engine.

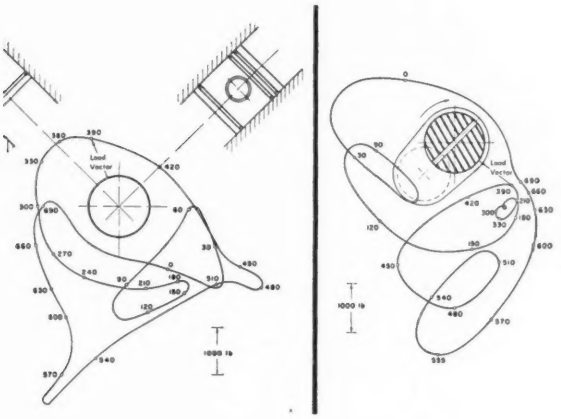


Fig. 2a (left) No. 4 Main Bearing Load—4000 rpm; Fig. 2B (right) No. 4 Main Journal Load—4000 rpm.

The trend toward engines of large bore and high compression ratio has resulted in a steady rise in unit bearing loads based on the maximum load encountered in the cycle. These changes have caused the gas loads encountered at low speed to exceed the inertia loads at high speed, which was generally the greatest load in older designs with longer stroke and smaller bore and compression ratio. The severity of load increase has been greater than would be judged on the basis of unit loads, since the ratio of bearing width to diameter has been reduced also. Currently, maximum unit loads on rod bearings of 3000 psi are common and projected designs approach 6000 psi. Main bearing loads are generally somewhat lower.

BEARING DETAILS. Frequently, the principal function of a bearing engineer is second guessing—deciding how to correct a difficulty encountered after the design is complete and a prototype has been built. This is not a surprising situation in view of the difficulties encountered in accurately predicting bearing loads or operating conditions before the engine is built and run. Oil temperature is such an unpredictable factor. It has a big effect upon bearing performance, but is very difficult to estimate. Distortion of the bearing housings and shaft deflections both greatly alter the permissible unit load which a particular bearing material will sustain under cyclic loading. These facts make any table of allowable unit loads for different bearing materials only an approximate guide. However, though the exact magnitude of bearing fatigue strength may not be accurately stated, the relative order of merit with respect to this characteristic may be. The following table is in order of increasing strength:

Thick babbitt (lead and tin)
Cadmium alloys
Copper-lead (35-40% lead)
Thin babbitt (.003 inch or less)
Aluminum alloys
Copper-lead (20-25% lead + tin)
Silver
Hard bronze

Fatigue strength is dependent upon bearing temperature, but for normal range of operation the above order remains the same.

Though there are many factors which cannot be evaluated in the design stage, the bearing designer can simplify his future problems by careful attention to many details which profoundly affect the performance of dynamically loaded bearings found in engines. Since little can be done about bearing size, it is essential that the best use be made of the space available. One of the most significant factors in this connection is the means of getting oil into the clearance space. What kind of grooving should be used? Where should the oil hole be located in the crankpin? Load diagrams both with respect to the bearing and the journal are essential in answering these questions.

OIL HOLES AND GROOVES. Consider the main bearings first. Unlike the rod bearings, the loads applied to the mains are not all alike, since the relative placement of the crankpin and the counterweights differ. It is, therefore, necessary to study each of the mains separately. A typical V-8 main bearing load diagram is shown in Fig. 2a. If the loads are found to be predominately in the lower half, as in diesel engines and high compression ratio short stroke gasoline engines, the effectiveness of the bearing can be greatly increased by omitting the groove in the lower half of the bearing normally used to supply oil to the rod bearings. In fact, in highly loaded bearings the load capacity of the bearing, based on the same film thickness, is approximately doubled when the groove is removed. Omission of the groove in the lower half makes it necessary to provide some alternate means of providing oil to the rod bearing or to operate with pressure feed to the rod only half of the time. A continuous supply to the rod may be obtained by drilling a hole through the diameter of the main journal. An intersecting hole then is drilled through the cheek to the crankpin.

The location of the oil hole in the main bearing journal is much more critical if the oil groove is omitted from the lower half of the bearing. It should be placed so that it is not in the area on the shaft which is subjected to high loads. This location may be determined from a load diagram drawn with respect to the journal, such as Fig. 2b. Lack of attention to this detail may result in the type of failure shown in Fig. 3. This shows a main bearing from a diesel engine which had been run on a dynamometer on a two-speed cycling schedule. It is the bottom half of the main bearing which is left ungrooved in order that the load capacity be as high as possible. The upper half of the bearing has a central oil groove and a diametral hole is drilled in

the journal from which the connecting rod is fed. The two half-moon shaped eroded areas correspond to the position of the oil hole in the shaft at the time of maximum load for each of the two speeds. Erosion was caused by the oil flowing back into this hole out of the highly loaded journal areas. Some investigators have reported that clean oil will not produce erosion even at extremely high shear rates. In engine service, however, there are always some foreign particles present in the oil. In addition, it is conceivable that the shear stress in the oil film might reach a value higher than the strength of the metal and thus remove it even though no particles were present.

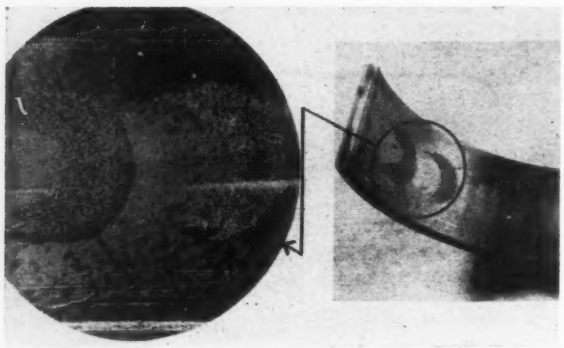
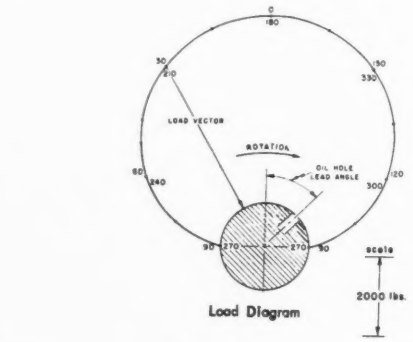


Fig. 3 Eroded main bearing.

Connecting rod bearings are generally not grooved and the placement of the oil hole in the crankpin is, therefore, an important consideration. It is obvious that oil will only be able to enter the clearance space in an area where the film pressures are low and the film thickness relatively large. Reference to the load diagram with respect to the crankpin, as shown in Fig. 1b, permits the selection of a suitable location. The inertia and rotating loads produce forces on the inside of the crankpin and their magnitude increases with speed. The load vector is in this area for a large portion of the complete 720° cycle. This is clearly not a desirable location for the oil hole. The outside of the pin is subjected to high loads resulting from gas forces—particularly at low engine speed. There is an area on the leading side of the crankpin where the oil hole is shown which is subjected to loads only on the compression stroke at lower speeds. The loads are low and are directed on this area for only a few degrees of crank rotation, as evidenced by the rapid change in direction of the load vector.

The importance of suitably locating the oil hole is demonstrated by the test results shown in Fig. 4. The load diagram applied by the test machine is similar to that found in a high speed engine with small gas loads. An oil hole lead angle of 0° corresponds to a hole on the inside of the crankpin in an engine operating under these conditions. The effect upon bearing life and the difference between the bearing temperature and the oil inlet temperature are shown. With oil hole placed in the load area no temperature equilibrium was ever attained and the bearings failed by overheating and scoring. As the oil hole is moved around out of the load area, the equilibrium temperature drops and the final failure which occurs is fatigue. Location of the oil hole 180° from the maximum load produced the longest fatigue life. For conventional engine designs the crankpin oil hole location shown in Fig. 1b is satisfactory.



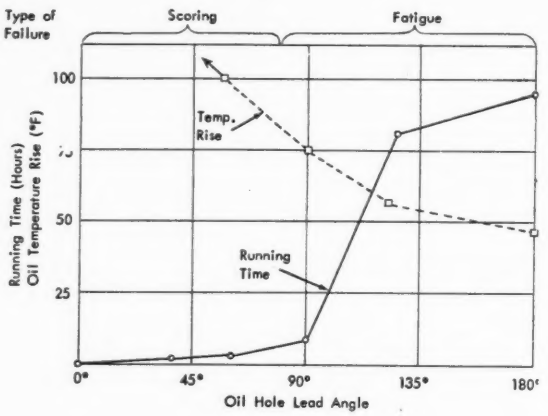


Fig. 4 Influence of oil hole location on bearing performance.

OIL SUPPLY. Thus far nothing has been said about oil pressure. Of course, the oil flow rate, and therefore bearing cooling is dependent upon the pressure, but more important is that the pressure at the main bearing be high enough to permit the oil to reach the center of the shaft against the centrifugal force produced by shaft rotation. If this pressure is not supplied, the rod bearing will be starved. The required pressure at the entrance to the passage from the main bearing groove to the rod is:

$$P = 4.6 \times 10^{-7} N^2 R^2$$

Where P is the pressure (psi), N is shaft speed (RPM), R is shaft radius (inches). This equation is based on a specific gravity of .9, which is approximately correct for oil.

This value does not take into account the pressure losses which result from oil flow through the oil gallery, the main bearing groove, and the oil passage in the shaft. The pump supply pressure must, therefore, be appreciably higher to assure an

adequate oil flow rate.

CLEARANCE. Oil flow rate is, of course, tremendously influenced by bearing clearance. In fact, it is roughly proportional to the cube of the clearance. Selection of the clearance is not entirely arbitrary. First, if the engine is for passenger car use, the clearance must be kept small enough to avoid objectionable noise. This maximum clearance is dependent upon the speed, the stiffness of the engine assembly, and the rate of pressure rise produced by the combustion chamber. These many factors make it difficult, if not impossible, to predict permissible clearance before the engine is built and run. A further restriction on the choice of clearance is the manufacturing tolerance required. In the case of passenger car engines the stack up of tolerances for housing bore, bearing wall thickness, and shaft diameter may permit a variation of .002 inch in the assembled clearance. Finally, wear occurring in service will further modify the design clearance. Care must be taken to provide a sufficiently large oil pump so that with the largest clearance encountered the oil pressure will not drop below that required to supply the rod bearings as noted above.

Two types of failure occur with dynamically loaded bearings that are never experienced when the load is fixed in magnitude and direction. The first of these failures is oil film breakdown caused by the specific nature of the loading. The second is fatigue of the bearing material as a result of cyclic oil film pressures. We will consider these in order.

DYNAMIC OIL FILMS. Oil film thickness in a unidirectionally loaded bearing is a function of the oil viscosity, journal speed, and the applied load. In addition, both the rotational speed of the load vector and cyclic changes in load magnitude influence the oil film thickness in engine bearings. The effect of these variables is not independent so that it is not possible to simply add their effects together. Analysis of the most general case, such as illustrated by Figs. 1 & 2, is involved, but the performance of bearings with a rotating load of fixed magnitude or with a fixed load direction and variation in magnitude only may be readily determined. The case in which the applied load of fixed magnitude rotates and the shaft also turns has been described by Stone and Underwood. Though the complete differential equations derived are difficult to handle, an extremely simple and useful relation was observed. This relation makes it possible to compare the dynamic load situation with static loading by means of a factor called "load carrying capacity." This term is defined as the ratio of the static load that must be applied to a bearing to produce the same eccentricity ratio as the dynamic loading under consideration. All factors, such as geometry and lubricant, remain constant in this comparison and only the loading is changed. From this definition, a dynamically loaded bearing with a load capacity greater than 1 would carry more load than a similar bearing unidirectionally loaded to produce the same eccentricity ratio. Similarly, load carrying capacities less than 1 indicate that the dynamic load has reduced the ability of the bearing to support load. The load carrying capacity of a bearing operating under these conditions can be shown to be:

$$LCC = (2N_L - N_J)$$

Where N_L is the angular velocity of the load vector and N_J is the angular velocity of the journal.

Some specific cases of this type of loading are shown in Fig. 5 along with their relative load capacity. An experimental test of this relation has confirmed the validity of the analysis. It will be seen that the loading frequency can reduce the capacity of the bearing to zero or increase it greatly. Both of these extremes may be encountered in engines. The practical importance of the relative load and journal speeds is evident from the fact that engines have been built which could not successfully be run at the designed load and speed as a consequence of bearing failure resulting from loss of oil film. No modification of bearing design or material will help in this case, since a stable oil film cannot be established except by changing the load diagram.

	Relative Load Carrying Capacity
1. Unidirectional Load. . . . . (Turbine bearing gear load bearing, etc.)	1
2. Rotating-Direction Load With Rotating Shaft . . . . . (Centrifugal bearing test machine bearing loaded by an eccentric weight)	1
3. Rotating-Direction Load With No Shaft Rotation . . . . . (Support for measuring friction torque in a bearing; outer oil film of a non-rotating floating bushing)	2
4. Critical Speed . . . . . (When the journal rpm is twice the frequency of load; special case is oil film whirl at twice resonant frequency of shaft whip)	0
5. Floating Bushing at Half Speed:	
Inner oil film	1/2
Outer oil film	1-1/2
6. Non-rotating Floating Bushing:	
Inner oil film (same as type 2).	1
Outer oil film (same as type 3)	2
7. High Capacity Bearing. . . . . (Load and journal revolving in opposite directions)	3

Fig. 5

There are many examples of engine bearings which operate with little or no shaft rotation and which are subjected to a reciprocating force. Such a bearing is exemplified by the piston pin bushing in a reciprocating engine. Here the bearing simply oscillates through an angle of perhaps 25° during each engine revolution. This motion between the

journal and the bearing is inadequate to provide a full oil film under the loads imposed, and the success of the bearing operation depends largely upon the character of the loading. The film thickness of such a bearing subjected to a load varying sinusoidally in magnitude is shown in Fig. 6. The journal position is plotted for three value of load. From this figure it will be seen that the minimum film thickness position lags the maximum load by 90°. As the load reverses, the journal travels first rapidly and then more slowly across the clearance space. This journal path is the result of the great variation in the stiffness of the oil film with eccentricity. Large increases in load are required to produce appreciable reduction in minimum oil film thickness.

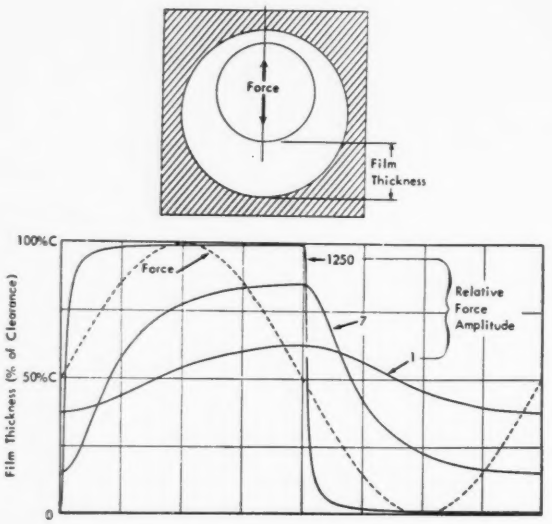


Fig. 6 Oil film thickness with sinusoidal loading and no rotation.

It should be noted that such a stable oil film is possible only if there is a reversal in load direction during the loading cycle. A load which varied in magnitude but was always applied in the same direction would cause the journal to travel continuously in one direction until the oil film was completely squeezed out and metallic contact occurred. The practical importance of this difference in loading is illustrated by Fig. 7. Piston pin bushings in the small end of the connecting rods from a four-stroke and a two-stroke engine are shown. The four-stroke piston pin bushing is simply a plain bore, while the two-stroke bushing is grooved to provide an opportunity for oil to contaminate the rubbing surfaces through the slight rotation of the rod in operation. No load reversal occurs in this two-stroke cycle preventing the formation of a "squeeze" film. Thus the operation of the two-stroke bushing is much more severe and necessitates more attention to the properties of the bushing material and the lubricant under boundary conditions. The wrist pin bushing, in the case of the four-stroke engine, is the most heavily loaded bearing in the engine but operates with very little difficulty.

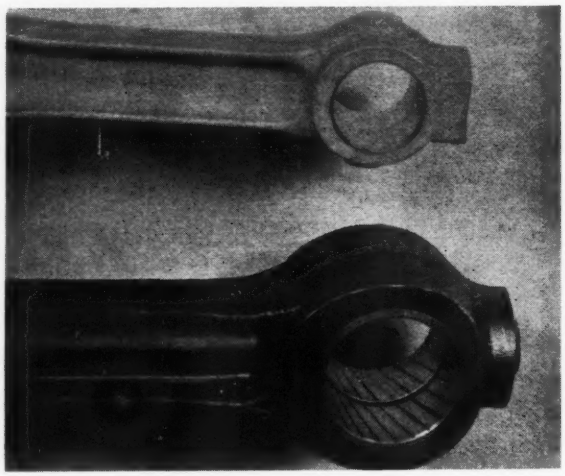


Fig. 7 Piston pin bushings for two types of load; (top) four stroke, (bottom) two stroke.

BEARING FATIGUE. Cyclic loads will also produce fatigue of bearing materials as a result of repeated change in oil film pressure from zero to its maximum value. Cracks appear in the bearing metal and pieces break out. The failure appears to be brittle though the material is ductile. This fatigue of bearing material is perhaps the most common type of failure under dynamic load. The cracking and breaking out of the bearing material occurs at a considerably lower load than is required to cause metallic contact and seizure. Since fatigue strength is a property of the material, it is obvious that more than hydrodynamics is involved in the solution of this problem.

Failure of bearing materials by fatigue was not recognized to be the result of this particular mechanism until rather recently. Such failures were instead generally blamed upon failure of the bond between the bearing metal and its backing. A typical example of fatigue in a thick babbitt bearing is shown in Figure 8. The edges of the cracks are sharp and the sides approximately vertical down close to the bond line. The crack then turns and progresses parallel to and just above the bond line. Several cracks finally meet permitting a small plate of bearing material to become loose. These pieces of bearing material then pass through the oil film and particles are frequently found embedded in other areas of the bearing.

A recent investigation of the stresses in a bearing composed of a relatively soft metal on a harder and stronger backing has cast light on many of the observed facts about bearing fatigue. This analysis indicates that the pressure distribution found in a bearing causes an "effective stress" in the bearing material which is very much less than the applied fluid pressure. It is for this reason that materials like babbitt, with a very low yield point, can sustain fluid pressures of many times this yield point. The

effective stress is dependent upon the elastic constants of the metal as well as upon the thickness of the bearing material. This last factor explains the observed fact that thin layers of babbitt have very much better fatigue life than do thick layers. The improvement in fatigue life resulting from the use of a thin bearing layer depends also upon the physical properties of the material at the operating temperature and load. Some of the harder bearing materials, such as silver, may have their smallest effective stress at a thickness somewhat greater than the optimum for babbitt. The thinnest layer of bearing metal does not always produce the greatest fatigue life.

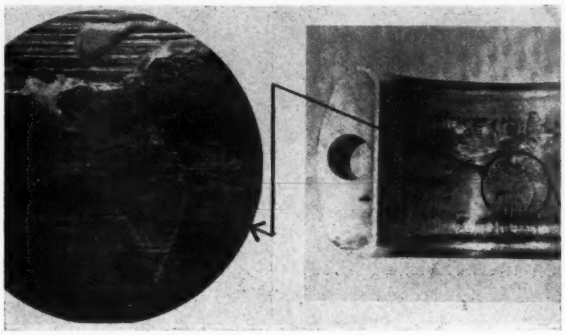


Fig. 8 Fatigued thick babbitt rod bearing.

The results of this analytical approach to the fatigue problem are currently being checked in a test fixture which applies a cyclic hydraulic pressure to a flat plate of bearing material bonded to a steel block. This test fixture is designed to simulate the pressure distribution of a bearing without the use of a rotating shaft. This pressure distribution is obtained by pumping oil between the flat test plate and a conical plate located close to it. A typical fatigue pattern in thick babbitt obtained in this test fixture is shown in Fig. 9. The character of the failure appears to duplicate that encountered in the bearings very satisfactorily. It will be some time, however, before a complete quantitative check of the analysis will be available.

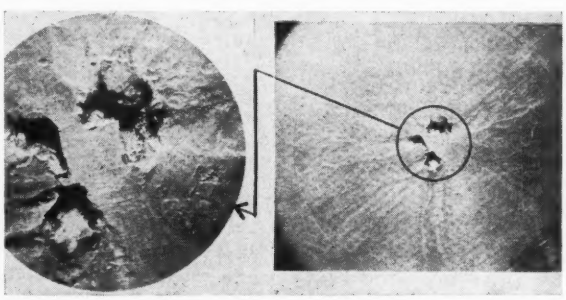


Fig. 9 Specimen fatigued by cyclic hydrostatic pressure.

Stress analysis simply indicates the magnitude of the stress in the bearing surface. It is then necessary to utilize bearing materials which have strength suitable for use under these conditions. During the past several years a number of stronger materials have been employed in bearings. These are represented by materials like copper-lead, aluminum, and silver. In addition to using these stronger materials, the trend has been towards thinner babbitt overlays which result in lower effective stresses. Thus, it has been possible to use the same material, babbitt, at continually higher loads. With the exception of fatigue strength, babbitt possesses all of the desirable characteristics of a good bearing material. It has good score resistance which, coupled with its soft nature, makes it tolerant of misalignment and dirt. These factors are essential qualities for a bearing material for use in engines. Composite bearings, such as copper-lead with a babbitt overplate or gridded silver, are attempts to combine the good features of babbitt with the strength of these harder materials. In general, the rating of these harder materials themselves with respect to score resistance and embedability are about in inverse order to their fatigue strength shown above.

It was mentioned earlier that it is not possible to provide a simple table of allowable unit loads for all bearing materials. We will now consider a few factors which affect bearing fatigue but which cannot be evaluated in the original design. All of these influence the peak pressures encountered in the bearing which are responsible for the fatigue. Perhaps the most obvious factor is the length/diameter ratio of the bearing. If a wide bearing of small diameter and a narrow bearing of large diameter are subjected to the same unit load based on projected area, the peak film pressures in the narrow bearing will be much higher. For this reason the omission of grooves in bearings greatly reduces fatigue though the unit load is only slightly affected.

Similar increased film pressures may result from geometric inaccuracy of journal or bearing produced either in manufacture or by wear. Polishing of crank journals with abrasive paper must be done carefully to maintain a cylindrical surface. When new bearings are placed on shafts which are worn, they may fatigue rapidly as a result of high areas produced in the previous running. Raised edges of nicks in the housing bore or foreign material between the bore and the bearing back may produce similar fatigued areas though they will be localized on the bearing surface over the defect. Sufficient allowance for endwise location of the bearings is essential to prevent the bearing interferring with the shaft fillet. Not only does this cause fatigue at the edge of the bearing, but the restriction in oil flow causes higher bearing operating temperatures and may result in seizure.

An effect similar to geometric inaccuracy is produced by faulty alignment of the axes of the bearing and journal. Fig. 10 illustrates two bearings which were run under exactly the same test conditions except that the shaft slope was varied. A slope of .002 inch/inch has caused severe fatigue damage to one while the other is unharmed. Though

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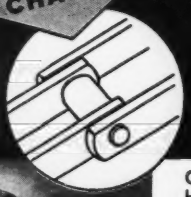

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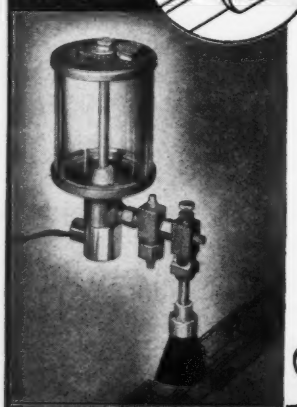

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engines are designed to have their bearing bores and journals parallel, shaft bending under load frequently causes edge loading which the bearing material must accept.

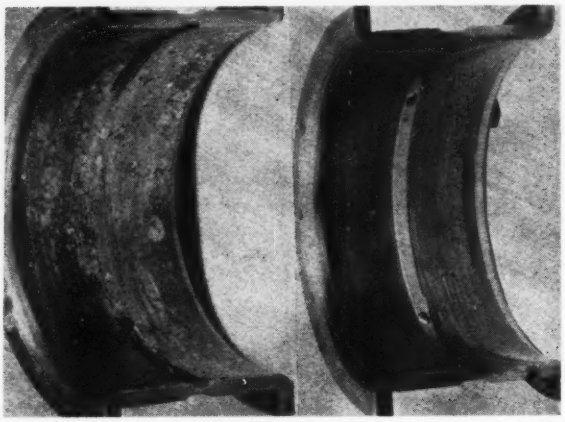

Fig. 10 Babbitt bearings run under same operating conditions except for shaft alignment; (left) shaft sloped with respect to bearing, (right) axes of shaft and bearing parallel.

Flexibility of bearing housings is another important factor. If the housing can deflect under load in a manner which will reduce film pressures, the fatigue life will be much greater than if rigid areas occur to locally increase pressures. Connecting rod bearings are an excellent example. If a conventional connecting rod is run in a test machine which loads alternately in one direction and then the other, it will be found that the fatigue occurs first directly under the rigid column section in the upper half of the bearing. Even though it has been subjected to the same total load, the cap half is not damaged. Attention must be given to the stiffness of the bearing support so that the load is distributed over as large an area of the bearing surface as possible.

LUBRICANT INFLUENCE—CORROSION. Since oil is a necessary accessory to the successful operation of bearings, it might be anticipated that lubricant characteristics would influence bearing performance. Though from the standpoint of hydrodynamics only the viscosity of the fluid is important, the practical utilization of mineral oils in internal combustion engines is complicated by many other properties. First, the temperatures encountered combined with crankcase contaminants originating in the combustion chamber may cause oil to oxidize and produce organic acids which will corrode many of the common bearing materials, such as cadmium and lead. This problem has been attacked in two ways—the development of bearing alloys which are corrosion resistant and the use of oxidation inhibitors in the oil to prevent formation of the undesirable acids. The extent of corrosion damage is influenced by both temperature and pressure. Bearings are often found which are corroded only in the high film pressure area immediately under the rod column section. The oil has been

further modified by the addition of detergents which keep finely divided material in suspension and prevent deposition on engine parts.

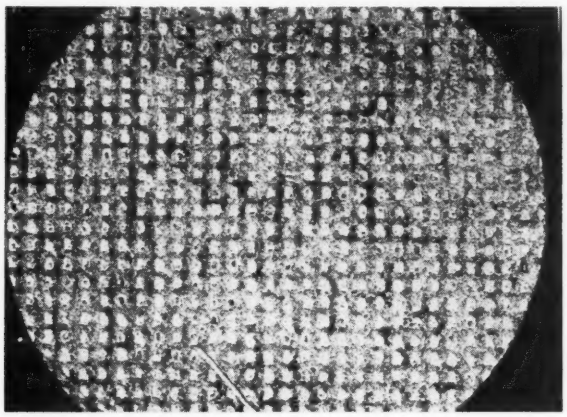

Fig. 11 Print from replica of silver grid bearing attacked by sulfur in oil—10X size.

SULFUR ATTACK. Along with the desirable results achieved by the use of these oil additives they may be the direct cause of other new problems. These stem from the particular chemical nature of the additive materials. The most common difficulty is sulfur attack of bearing materials, such as silver and copper. Sulfur compounds are an especially common type of oil additive, and unless the sulfur is tightly bonded in the chemical compound it may directly attack these bearing materials. Elevated temperatures accentuate the problem. Water as a contaminant in the crankcase has also been found to similarly accelerate sulfur attack in some cases. Fig. 11 is a print from a replica of the surface of a silver grid bearing which has suffered such damage. In this case the lead is found raised above the silver which has been corroded away. Oils which are noncorrosive or only mildly corrosive may become extremely active at low temperatures in the presence of water. Fig. 12 is a silver thrust washer which has suffered sulfur corrosion. This washer was overplated with babbitt. The 500 X section through the edge of the corroded area shows how the babbitt protects the silver as long as it remains intact. The silver has been removed back under the babbitt overplate. Any breaks in the babbitt, such as scratches caused by dirt or wear in service, will expose the silver to attack.

Different bearing metal alloys exhibit varying degrees of resistance to sulfur corrosion. This has frequently been observed in the case of piston pin bushings. Copper-lead-tin bushings have been badly damaged by an oil which did not attack a bronze containing a small amount of zinc. Similar resistance to corrosive damage is possessed by some aluminum bronze alloys.

CORROSION FATIGUE. Corrosion fatigue is another type of damage which is in reality a function of the oil used, but which is often not recognized as due to this factor. This is a natural error

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since the appearance of the final failure is exactly like a normal fatigue failure. The effect of the lubricant is made evident by first determining the fatigue life of a bearing with an oil that does not promote corrosion fatigue and then running a duplicate test with an oil that does. A large difference in bearing endurance will be found. Copper-lead is especially sensitive to this acceleration of fatigue. It should be noted that though this phenomenon is called corrosion fatigue, there is no corrosion in the sense that either copper or lead is removed by the oil. Again it is possible to reduce the sensitivity of the copper-lead. One method of accomplishing this is to use a lead-tin overplate. The effectiveness of this cure has been indicated in a bearing test machine. With an oil which promotes corrosion fatigue a copper-lead bearing was found to have a life of about 18 hours. A similar bearing to which a .0006 inch overplate had been applied did not fail until after 100 hours under the same operating conditions. The protection is lost if the overplate is removed in service.

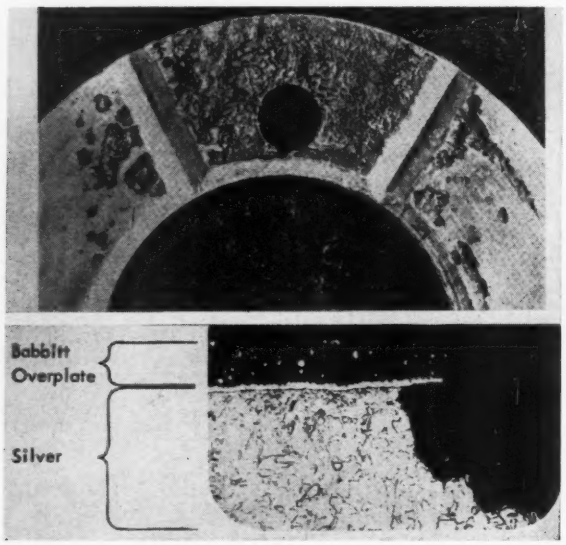

Fig. 12a (top) Overplated silver thrust washer corroded by sulfur; Fig. 12b (bottom) Section through edge of corroded area—500X size.

BEARING & SHAFT WEAR. The principal cause of wear in rod and main bearings is foreign material carried into the clearance space by the oil. Generally this material is embedded into the softer bearing surface and wears the shaft through the thin oil film. The wear occurs slowly and in a manner similar to lapping. The material is removed quite uniformly across the surface of a bearing fed by an oil groove like a main bearing—with the exception of the area immediately adjacent to the edge of the groove. In the case of rod bearings, the wear is often greatest on the half nearest the main bearing from which the oil is supplied. This is the result of the centrifugal force applied to the oil as it passes through the oil hole in the shaft. This

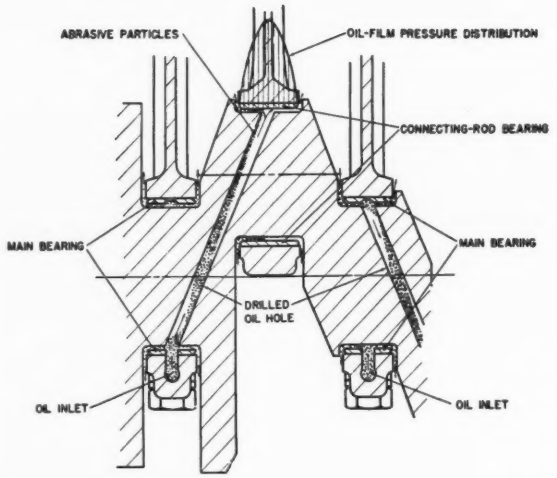

Fig. 13 Centrifuge effect of oil hole in crankshaft.

process is shown in Fig. 13. The oil film pressure causes the dirt to go out of the bearing predominantly in one direction. Damage of this type can only be prevented by keeping the dirt out of the bearings. Attention to this detail should be given in the engine design stage so that provision may be made for effective filtration. It is strongly recommended that full flow filtration be used to prevent any large particles from getting to the bearings. Though the usual result of dirt in the oil is wear

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which occurs slowly over a long time, sudden addition of a large amount of dirt to a bearing may cause immediate seizure after a short period of rapidly rising bearing temperature. Again, the only real cure is keeping this material out of the bearing, but bearing materials differ widely in their ability to handle dirt. In general, the softer materials, such as babbitt, excel in this respect and hard materials like silver will tolerate only very small quantities of dirt.

CONCLUSION. Though all of the foregoing material has been concerned with oil film bearings, such as are found on the crankshaft of the engine, it is important to be aware of the many bearing surfaces in an engine which operate under boundary lubrication conditions. Piston rings and cylinder bores, cams, and valve lifters, rocker arms and shafts all depend upon boundary lubrication for satisfactory performance. Consequently, they are all sensitive to specific oil composition. A complete discussion of the factors involved in these locations is outside the scope of this paper, but the bearing designer should be aware of possible problems.

The current freedom from bearing problems in passenger car engines is a tribute to the progress made in materials and designs in the past several years. However, bearing requirements can be expected to become more severe as specific power output and compression ratio are further increased. Under these circumstances more attention must be paid to means of improving fatigue resistance without sacrifice of embedability and to bearing design which will most efficiently use the space available.

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(New Products, from page 265)

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CAPACITIVE LEVEL INDICATOR. Capable of providing a continuous indication of level for a wide range of materials in storage or process, the new "Belmont Level Indicator" utilizes a capacitive-type sensing-element probe with no floating or moving parts nor contact-making electrodes. Operable over a temperature range of 500 to — 425 F., and from high vacuum to 100,000 psi. Various probe lengths, materials, and designs are available to cover different ranges in the rise and fall of materials, the character of substances encountered, and such special problems as the non-entraining requirements of relatively viscous materials. Standard instruments provide for a ten-foot separation between the probe and the equipment housing, however the interconnecting coaxial cable can, if necessary, be as long as 1,000 feet. Design is such as to permit inexpensive addition of remote indicating instruments over telemetering circuits of almost unlimited length. For further information write: Thermo Instruments Co. (LE10/5), 1310 County Rd., Belmont, Calif.

TEFLON BELLOWS. Chemically inert, corrosion and heat resistant bellows made from duPont Teflon are now available with companion flanges conforming to American Standard bolt hole circles and numbers for pressures up to 125 psi; they can be supplied to fit any pipe size from ½ through 12 in. Since they retain formation at full free length, they are subjected to minimum residual and working strain — they expand and contract in either direction with equal freedom of motion. Operating temperature range is from — 94 to + 500 F. Bellow flanges are of French-type gasket construction to provide more efficient sealing at lower flange pressure; gasket insert material is asbestos or synthetic rubber. Typical applications include vibration dampeners, expansion joints and connectors; they are available in a wide range of designs suitable for metering pumps, pressure accumulators, batching scales, hydraulic systems and others. For complete information, write: Crane Packing Co. (LE10/5), 1800 Cuyler Ave., Chicago 13, Ill.

LOW TEMPERATURE BATH & CIRCULATING SYSTEM. "Lo-Temptrol," a low temperature bath and circulating system designed as a convenient means of reducing temperature rapidly and holding it at a precisely controlled

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point, is equipped with a combination stirrer and pump and is available in two sizes with two temperature ranges. The small, 15 gallon bath has a range from room to — 30 C.; the large, 45 gallon bath, from room to Zero C. A microset thermoregulator controls temperatures to ± 0.1 C. near room temperature, to ± 0.01 C. at lower temperatures. Bath capacities are ample to permit simultaneous use as bath and circulating system, without affecting the sensitivity of control. The bath chamber is all stainless steel, including the rack above the cooling coil. Write for Data Sheet No. 11297: Precision Scientific Co. (LE10/5), 3737 W. Cortland St., Chicago 47, Ill.

PERMATEX SEALANT. Development of a new industrial sealing compound to solve the costly problem of leakage of cutting and cooling oil in machine tools has been announced. Known as "Sealant No. 1372," the compound is a liquid which extensive tests have shown to be impervious to high temperatures and resistant to the new synthetic cutting and cooling oils used with modern lathes, drills, and other machine tools. Seepage and leaks of these cutting oils into tool gear boxes dissolves lubricants and causes frequent and highly expensive shut-downs of production tools in order to re-lubricate gears. Use of the new sealant prevents such leakage and keeps productive equipment operating for longer periods. For additional details, write: Permatex Co., Inc. (LE10/-5), Brooklyn, N. Y.

GEAR TESTING MACHINE. Simulated service fatigue tests of gears under the loads and at the speeds encountered in normal operation are provided in a special testing machine developed by Sonntag Scientific Corp., Div. of Baldwin-Lima-Hamilton Corp., in co-operation with International Harvester Co. Test gears can be subjected to torques up to 30,000 in-lb. with a maximum tooth load of 14,400 lb. Pinion speed can be varied between 750 and 2500 rpm, to transmit a maximum of 1200

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Book Reviews

STATISTICAL ANALYSIS IN CHEMISTRY AND THE CHEMICAL INDUSTRY, by C. A. Bennett & N. L. Franklin; John Wiley & Sons, Inc., 440 Fourth Ave., New York 16, N. Y., 1954, 724 pages, price $8.00.

This book is a valuable addition to the library of anyone interested in the interpretation of experimental or of analytical results. The authors state "Interest in the application of statistical methods to all types of problems has grown rapidly since the late 1920's. Although the initial development of these methods was, to a large extent, connected with biological and agricultural experimentation . . . they are at present being applied in almost every field of scientific investigation."

The text starts with the basic fundamental theory of statistics and develops each topic quite completely, giving the theoretical background and derivations of analytical methods along with computational procedures. The first five chapters develop the basic principles of statistical inference, while the remaining six contain specialized methods which have been found to have the greatest utility in industrial applications. Numerous illustrative examples are given, along with elaborate tables and appendices; a large number of references are included at the end of the book for those interested in source material in this field. The user of this text should have some familiarity with higher mathematics through the calculus in order to fully comprehend the developments.

B. H. Jennings
Northwestern University

TECHNIQUES OF PLANT MAINTENANCE & ENGINEERING—1954, Proceedings of the Plant Maintenance & Engineering Conference; Clapp & Poliak, Inc., 341 Madison Ave., New York 17, N. Y., 291 pages, price $7.50.

Regarded in engineering circles as the most important statement of current factory maintenance problems, this book is the most extensive ever published in the series—it contains texts of papers read by 24 authors, and direct answers to more than 1,300 questions.

The two principal sessions were devoted to "Planning & Scheduling Maintenance Work," and "Maintenance Cost Control & Budgeting." Other sessions covered preventive maintenance; training people for maintenance work; organization, policies and procedures for effective maintenance; sanitation; work measurement, standards and incentives; getting management approval for maintenance programs; handling maintenance labor; corrosion; hospitals, medical departments and first aid rooms; and waste disposal.

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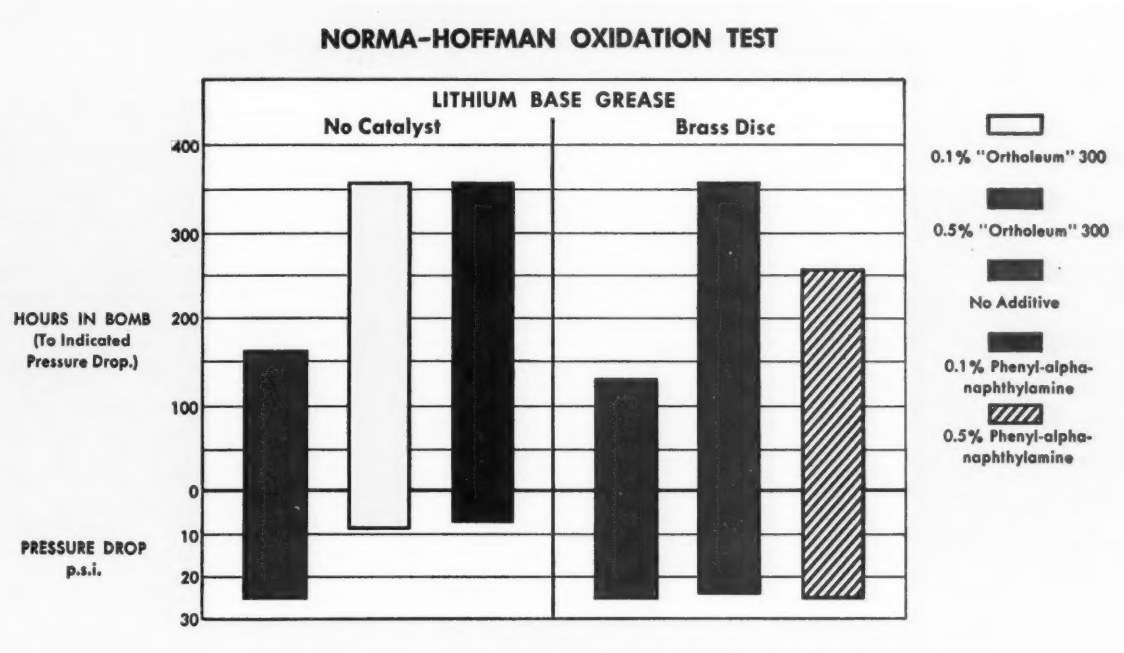

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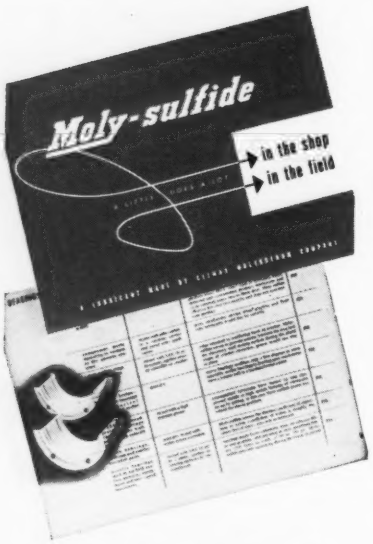

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summary of 20 roundtable discussions (each of which ran for five hours) devoted to petroleum refineries, sheet metal plants, steel mills and textile mills, paper mills and paper product plants, breweries and distilleries, and food processing and packaging plants. Roundtables, other than those devoted to specific industries, covered maintenance of plant buildings; electrical distribution systems; electronic controls; use of handling equipment in maintenance work; maintenance of materials handling equipment; lighting equipment; lubrication; plants having less than 25 maintenance employees; maintenance storekeeping; mechanical service equipment; power plants, and janitor work.

Personals

Dr. S. K. Talley, Chemist of Shell Development Co., Emeryville, Calif., and Editor of **Lubrication Engineering,** has been given a one-year assignment to work on industrial lubricants at the home office of Shell Oil Co., New York City.

During the combined Iron & Steel Exposition and Annual Convention of the Association of Iron & Steel Engineers, Cleveland, September 29, **N. I. Whiteley,** American Steel & Wire Div., and **W. M. Schuck,** Armco Steel Corp., presided as Chairmen of the Lubrication Session. ASLE members who presented papers included: **C. R. Schmitt,** E. F. Houghton & Co.; **W. A. Magie, II,** Magie Brothers Oil Co.; and **C. D. Thayer,** Socony-Vacuum Laboratories.

Acheson Industries, Inc. has introduced a new trademark as a common denominator for four affiliated manufacturing companies: Acheson Colloids Co.; Acheson Colloids, Ltd.; Acheson Dispersed Pigments Co.; and Gredag, Inc. Acheson Colloids Co., Port Huron, Mich., has announced the opening of a Rochester, N. Y. sales headquarters (545 Powers Bldg.) with **E. A. Lampman,** Service Engineer, in charge. Gredag, Inc., Niagara Falls, N. Y., has announced the election of **H. A. Acheson** as President, **W. Ewart** as Vice-President, and **J. C. Sprague** as Secretary-Treasurer, with **M. W. Reynolds,** Acheson, and Sprague as Directors.

C. A. Arents, Co-ordinator of Research, Illinois Institute of Technology, has been named Secretary of the National Conference on Industrial Hydraulics to be held Oct. 14 & 15 at the Sheraton Hotel, Chicago, Ill.

Standard Oil Co. (Ind.), Chicago, Ill., has announced new positions for several sales executives: **J. W. Ross,** Assistant General Manager-at-Large; **W. H. Miller,** Ass's Gen. Mgr., Sales Promotion & Advertising; **J. M. Hamer,** Ass't. Gen. Mgr., Administrative; **C. H. Lamoreaux,** Ass't. Gen. Mgr., Gasoline & Fuel Oils;

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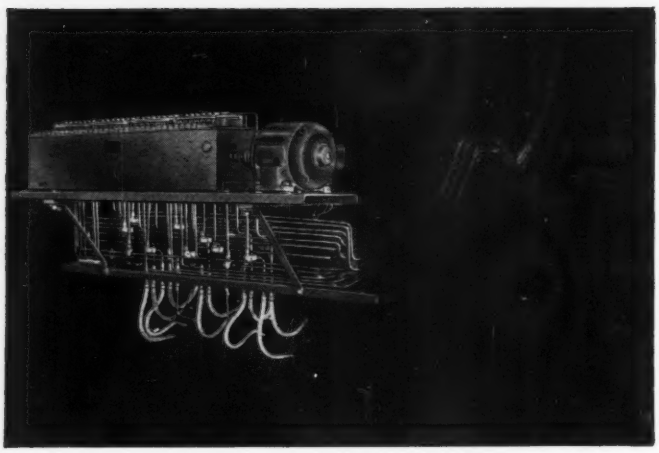

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and **G. T. Dougherty,** Ass't. Gen. Mgr., Industrial Sales & Technical Service.

A. Sonntag has resigned as Manager of Sonntag Scientific Corp., Div. of Baldwin-Lima-Hamilton Corp., to devote his time more exclusively to The Alpha Corp., Greenwich, Conn., of which he is President, but will continue in an engineering consultant capacity for Baldwin-Lima-Hamilton.

Obituary: **H. I. Glashoff** (ASLE Baltimore Section), American Oil Co., Cheverly, Md., August 17.

(Current Literature, from page 261)

light but sturdy Trabon filters constructed of high tensile cast aluminum with steel sump and tie rod, and built to withstand 1000 psi working pressure and 1800 psi test pressure.

Model S-216-C Series of Air-Driven, High Pressure, Hydraulic Pumps. Sprague Engineering & Sales Corp. (LE10/5), Gardena, Calif. (4 pages)

Lists eight models with discharge pressures from 1,100 to 33,500 psi and describes operation of units in high-pressure, low-volume service such as hydrostatic testing, hydraulic presses and other hydraulic machines for clamping, riveting, crimping, shearing and punching.

Ask Anyone in Industry, Form 34-22. Alemite Div., Stewart-Warner Corp. (LE10/5), 1828 W. Diversey Blvd., Chicago 14, Ill. (24 pages)

A lively, fast-paced "comic book," cartooning and lampooning all of the grease-splotched misadventures encountered with old-fashined industrial lubrication practices. With no page containing more than eleven words of type matter, its fast reading pages scramble distracting sweater girls, frustrated grease monkeys, finger-losing oilers (who forgot to turn off the machine before squirting), and a host of other delineations of "characters," foibles and fallacies all too familiar to people in industry. It comes out in the

final few pages that the use of Alemite "Accumeter" measuring fittings—connected by tubing to a lubrication pump controlled by an automatic timer — eliminates all of the down-time, safety hazards, bearing failures, lubricant waste and other negatives inherent in outmoded types of lubrication.

Hydrofol Products. Archer-Daniels-Midland Co. (LE10/5), 2191 W. 110th St., Cleveland, Ohio (40 pages)

Technical catalog of fatty chemicals, designed to streamline reference and specification. Written from a functional standpoint, it is subdivided into four major product classifications: fatty acids, glycerides, sperm oils, and fatty alcohols; each section is then classified into reaction data, specifications, composition and application information. Working charts and test definitions give the catalog continuous desktop utility.

Centrifuges for Oil Processing, Booklet 205. Centrico, Inc. (LE 10/5), Englewood, N. J. (16 pages)

In view of the growing use of low-cost heavy fuel in diesel engine operation, for power stations, aboard ship, railroads, and wherever diesels are used for power operation, this publication gives performance data of Westfalia's three oil separator models for purifying fuel and lube oils, with special emphasis on No. 6 residual oil, and describes Westfalia's new hydraulic lifter and exchangeable bowl liners that cut cleaning down-time 60 percent.

Lubrication Properties & Industrial Applications of Pure Molybdenum Disulfide. The Alpha Corp. (LE10/5), 179 Hamilton Ave., Greenwich, Conn. (4 pages)

Presented in this paper is a new concept of lubrication by solids, according to which extreme pressure or boundary lubrication without the participation of solid lubricants is practically impossible. It is held that thin oxide films, which act as lubricants and reduce friction by substantial amounts, form on most metals almost immediately after cutting or shearing but can be improved upon by alloying or supplementing them with more highly qualified solid lubricants. The role played by molybdenum disulfide in filling this need for improvement is the subject discussed in detail in this technical paper. (Presented before ASLE Fort Wayne Section.)

(Patent Abstracts, from page 255)

taining an average of from about 2.0 to 2.25 total methyl and phenyl groups per silicon atom and (2) from 5 to 15%, by weight, based on the weight of the aforesaid methyl phenylpolysiloxane of an organometallic salt selected from the class consisting of copper and aluminum sebacates.

Mineral Oil Compositions of Suppressed Foaming Tendency, Patent No. 2,682,508, by K. L. Godfrey, assignor to Monsanto Chemical Co.

A mineral lubricating oil composition comprising a major proportion of a hydrocarbon oil containing a small amount of an alkaryl phosphorus ester additive and an amide of a lower aliphatic carboxylic acid in small amount sufficient to decrease the normal foaming properties of the composition.

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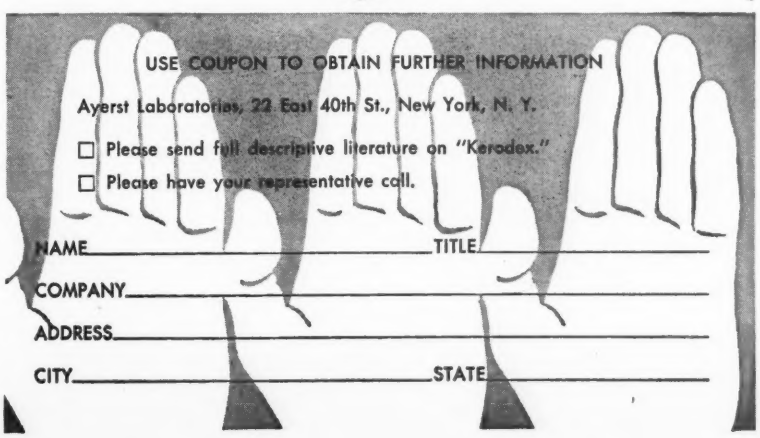

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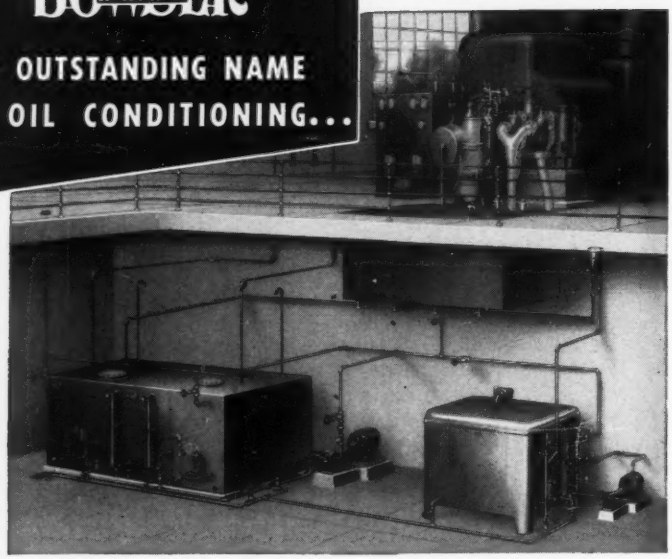

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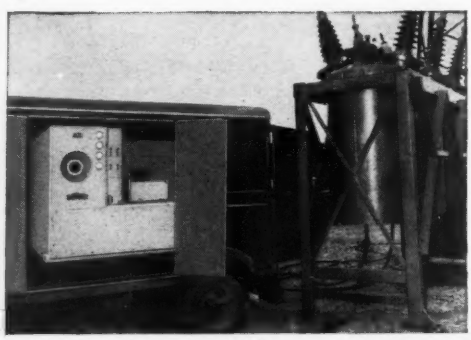

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Lubricants, Patent No. 2,682,523, by S. K. Talley & T. E. Reamer, assignors to Shell Development Co.

An aqueous emulsion metal-working lubricating concentrate composition containing from 3 to 50% of a halogenated high melting residue hydrocarbon wax containing 20-60% halogen, the hydrocarbon wax having a melting point of at least 160 F., a molecular weight above 475, and consisting of at least 60% normal straight-chain hydrocarbons and the remaining constituents being high melting isoparaffinic and naphthenic hydrocarbons; from about 1% to about 25% of an ester selected from the group consisting of an ester of a polyhydric alcohol and a carboxylic acid and a polyalkylene glycol ether of a partial ester of a polyhydric alcohol and a carboxylic acid; from about 0.7% to 10% of an organic nitrogen-containing compound having a hydrocarbon chain of from 8 to 18 carbon atoms selected from the group consisting of an aliphatic amine, and an aliphatic amide; and the balance being water.

Extreme Pressure Lubricants, Patent No. 2,683,691, by R. E. Thorpe & E. R. Bell, assignors to Shell Development Co.

A lubricating composition of matter comprising essentially a major amount of a lubricating oil and a minor amount sufficient to impart extreme pressure properties to said medium of a salt.

Lubricating Oil Containing a Reaction Product of P_2S_5-Terpene and 2.4.6.-Trialkylphenol, Patent No. 2,684,334, by J. G. Ryan & G. L. Hayes, assignors to Shell Development Co.

A non-ash containing lubricating oil composition for aviation engines consisting essentially of a mineral lubricating oil having incorporated therein from about 0.1 to 5% each of a reaction product of P_2S_5-terpene and of 2, 4, 6-trialkylphenol, at least one of the alkyl radicals ortho to the hydroxyl group being a tertiary alkyl radical.

Silicone Lubricant Containing Trialkyl Phosphate, Patent No. 2,684,336, by D. H. Moreton, assignor to Douglas Aircraft Co.

The liquid lubricating composition consisting essentially of a liquid dimethyl silicone polymer lubricant and a sufficient proportion of trialkyl phosphate in which the alkyl groups have from about 3 to 10 carbon atoms per alkyl group to increase the anti-seize lubricating property of said composition over that of said dimethyl silicone polymer.

Lubricants, Patent No. 2,684,943, by E. R. Baker, assignor to Continental Oil Co.

In an organic composition having hydrogen sulphide in solution containing a minor but effective amount of an organic epoxy compound to inhibit the

malodorous characteristics caused by the evolution of hydrogen sulphide from said composition, the improvement which consists in including in said composition, as a catalyst cooperating with the epoxy compound to activate the same, a minor amount of a metallic salt of an organic sulphonic acid.

Lithium Polyorgano Siloxane Polymer Grease Compositions, Patent No. 2,684,944, by S. J. Zajac, assignor to Standard Oil Co.

A lubricant grease composition comprising a major proportion of a liquid polyorgano siloxane, from about 1% to about 20% of a lithium soap of a fatty acid, and a free basic compound lithium soap content of said composition.

Hydrocarbon Wax Deoiling, Patent No. 2,685,557, by W. J. Dominicus van Dijck & W. M. Mazee, assignors to Shell Development Co.

The process of producing a substantially oil-free wax from a wax-containing mineral oil which comprises: forming a solution of said mineral oil with a first liquid consisting essentially of a ketone dewaxing solvent; cooling said solution to a temperature sufficient to precipitate solid wax; separating therefrom solid wax containing an oil-containing liquid comprising said first liquid dewaxing solvent together with oil dissolved therein; washing said separated solid wax with a second liquid characterized by having an absolute viscosity at the temperature of contacting at least 1.5 times greater than the absolute viscosity of said oil-containing liquid and further characterized by being a solvent for oil and substantially a non-solvent for wax at the temperature of contacting, the washing being effected under conditions to prevent the dissolving of substantial amounts of solid wax in said second liquid, and recovering a solid wax substantially free of said oil-containing liquid.

Treatment of Acidic Oils, Patent No. 2,685,558, by L. P. Hodges, assignor, by mesne assignments, to Standard Oil Development Co.

A method for neutralizing a petroleum lubricating oil which has been treated with sulfuric acid under conditions to form a hydrocarbon layer and an acid layer, which comprises separating a hydrocarbon layer containing acidic bodies selected from the group consisting of naphthenic, sulfonic and inorganic acids from the acid layer, contacting the hydrocarbon layer at a temperature in the range between 40 and 210 F. and at a pressure sufficient to maintain a liquid phase with a treating reagent comprising an aqueous solution of an alkali metal hydroxide, an alkali metal salt of an inorganic acid, and an alkali metal salt of a preferentially water-soluble aromatic sulfonic acid, said treating reagent having a specific gravity no less than 1.1 under conditions to remove acidic bodies selected from the group consisting of naphthenic, sulfonic and inorganic acids from said hydrocarbon layer, separating the treating reagent from said contacted hydrocarbon layer and recovering said contacted hydrocarbon layer substantially free of said acidic bodies.

Lubricant Composition Containing Alkyl-Mercaptoalkyl Phosphites, Patent No. 2,685,567, by D. Harman, assignor to Shell Development Co.

An improved lubricant comprising a liquid hydrocarbon lubricant base, having incorporated therein, in an amount sufficient to stabilize said base against oxidation deterioration, an organic ester of phosphorous acid in which the organic radicals which replace hydrogen atoms of the hydroxyl groups of said acid have the formula —R—X—R'— in which R and R' each are saturated non-aromatic hydrocarbon radicals containing from 1 to 15 carbon atoms, X is an element selected from the group consisting of sulfur and selenium, and from about 0.01% to about 2% of an oil-soluble detergent.

Process for Producing a Molded Solid Molybdenum Disulfide Lubricant Member, Patent No. 2,686,155, by H. B. Willis & H. S. Hopkins, assignors to Westinghouse Electric Corp.

In the process of producing molded solid lubricant members, the steps comprising intimately admixing from 79.9 to 97.5 parts by weight of a solid lubricant composed of finely divided molybdenum disulfide, from 20 to 0.5 parts by weight of a thermosettable resinous molding powder selected from the group consisting of urea-aldehyde resin, melamine-aldehyde resin, polyester resin, and phenol-aldehyde resin admixed with one methylene containing compound of the group consisting of paraformaldehyde and hexamethylene tetramine, from 0.1 to 2 parts by weight of an alkaline stabilizer selected from the group consisting of the alkaline earth metal oxides, hydroxides, carbonates and ethoxides, and molding the mixture under heat and pressure to cause the resinous molding powder to react to a thermoset resin thereby producing the solid member, the stabilizer cooperating to prevent development of acidity in the solid lubricant.

Preparation of Stable Molybdenum Disulfide Lubricants, Patent No. 2,686,156, by C. E. Arntzen & P. E. Dupler, assignors to Westinghouse Electric Corp.

In the process of preparing a stable powdered solid lubricant from finely divided, purified molybdenum disulfide, the steps comprising thoroughly admixing 100 parts by weight of the finely divided, purified molybdenum disulfide with sufficient parts by weight of an organic liquid composition so that the parts by weight thereof divided by its density is at least 30, the organic liquid

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Esters From Sperm Oil, Patent No. 2,686,191, by E. Grunthal & L. Mannes, assignors to Henkel & Cie, G.m.b.H.

Method for the production of fatty alcohol esters of unsaturated fatty acid from sperm oil having an acid value below 5, which comprises heating such oil in the presence of water at a temperature sufficient to split the glycerides in said sperm oil and insufficient to appreciably split non-glyceride esters in said sperm oil, and recovering practically neutral fatty alcohol esters of unsaturated fatty acids from the reaction product.

Oil Compositions Containing Wax-Alkylated Styrene-Isoprene Copolymers, Patent No. 2,686,759, by J. J. Giammaria, assignor to Socony-Vacuum Oil Co., Inc.

A mineral lubricating oil containing a minor amount, sufficient to depress the pour point of said oil, of the product formed by alkylating a styrene-isoprene copolymer, comprising approximately 75 weight per cent of styrene and the remainder isoprene, with an amount of chlorinated paraffin wax containing from 10 to 12 weight per cent of chlorine, to supply to the reaction from 2 to 4 atomic proportions of chlorine per mol of styrene in said copolymer in a polychlorinated hydrocarbon solvent having a boiling point of from about 80 C. to about 100 C., at the reflux temperature of said solvent, for a time period of from 5 to 7 hours and in the presence of approximately 3 per cent by weight of aluminum chloride as catalyst based on the combined weight of the chlorinated wax and copolymer used in the reaction.

Hydraulic Fluids and Lubricating Compositions, Patent No. 2,686,760, by F. J. Watson, assignor to Shell Development Co.

A composition comprising essentially from about 50% to about 92.5% by weight of a normally liquid trialkyl phosphate, said ester having at least 18

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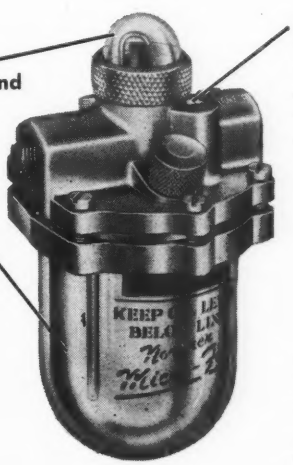

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carbon atoms per molecule, from about 7.5% to about 60% by weight of a mineral oil bright stock having a viscosity of between about 1250 and 11,000 SUS at 100 F., said oil containing less than about 15% by weight of aromatic hydrocarbons and said composition containing from about 0.25% to about 5% by weight of an alkaline earth metal salt of a condensation product of formaldehyde with alkyl phenols the alkyl groups thereof containing from 4 to 12 carbon atoms, and 0.25-5% by weight of a naphthylamine.

Section News

Notice to Section Secretaries: To insure early and adequate coverage of your Section's activities, all Section News copy (including Summaries of papers presented, Election of Officers, Photographs, etc.) must be received by the ASLE National Office 1½ months prior to he month of LE publication.

BALTIMORE, June Meeting. Election of Officers (see ASLE Directory. Submitted by W. W. Witmer, Vice-Chrmn.)

CHICAGO Program Schedule. Oct., Stump the Experts; Nov., Petroleum Lubricating Greases; Jan., Cutting Oils; Feb. Ladies' Night; March, Oil Filtration; April, Solid Lubricants; May, Hydraulic Oil; June, Golf Party. (Submitted by W. P. Green, Vice-Chrmn.)

CLEVELAND, April meeting. Election of Officers (see ASLE Directory. Submitted by R. R. Slaymaker, Retiring Chrmn.)

CONNECTICUT, June meeting. Election of Officers (see ASLE Directory. Submitted by P. Faxon, Sec'y-Treas.)

KINGSPORT, June meeting. Film presentation of "Smoothing the Way," courtesy of Esso Standard Oil Co. (Submitted by J. E. Fleenor, Sec'y-Treas.)

LOS ANGELES, April meeting. Election of Officers (see ASLE Directory), together with an address by R. Q. Sharpe, Socony-

Vacuum Oil Co., entitled "Industrial Hydraulic Systems" dealing with their selection, operation and maintenance. (Submitted by R. S. McCord, Retiring Sec'y.)

MONTREAL, June meeting. Election of Officers (see ASLE Directory. Submitted by F. C. Purvis, Chairman.)

PITTSBURGH, February meeting. Election of Officers (see ASLE Directory).

November. (See Lube Lines announcement concerning Section-sponsored Conference on Lubrication to be held Nov. 10 & 11. Submitted by S. S. McNary, Sec'y.)

Coming Events

OCTOBER

12-15 American Chemical Society Chicago Section (8th National Chemical Exposition), Coliseum, Chicago, Ill.

12-15 National Association of Corrosion Engineers (annual South Central Region conference), Adolphus Hotel, Dallas, Tex.

17-20 American Institute of Mining & Metallurgical Engineers (Fall meeting, petroleum division), Plaza Hotel, San Antonio, Tex.

17-23 American Society for Metals (national metal cong. & national metal exposition), Statler Hotel, Cleveland, Ohio.

18-19 Joint ASME-ASLE Conference on Lubrication, Lord Baltimore Hotel, Baltimore, Md.

18-20 Society of Automotive Engineers (national transportation meeting), Sheraton Plaza Hotel, Boston, Mass.

19 American Society of Safety Engineers (annual meeting), Conrad Hilton Hotel, Chicago, Ill.

25-26 National Lubricating Grease Institute (annual meeting), Mark Hopkins Hotel, San Francisco, Calif.

26-27 Society of Automotive Engineers (national diesel engine meeting), Statler Hotel, Cleveland, Ohio.

28-29 American Society of Mechanical Engineers (fuels-AIME coal conference), William Penn Hotel, Pittsburgh, Pa.

28-29 Society for Advancement of Management (annual Fall conference), Statler Hotel, New York, N. Y.

NOVEMBER

1-3 American Institute of Mining & Metallurgical Engineers (Institute of Metals Div. meeting), Sheraton Hotel, Chicago, Ill.

1-5 American Society for Metals, International Amphitheatre, Chicago, Ill.

8-11 American Petroleum Institute (34th Annual Meeting), Conrad Hilton Hotel & Palmer House, Chicago, Ill.

10-11 ASLE Pittsburgh Section (Conference on Lubrication), Pittsburgh, Pa.

15-17 American Standards Association (annual meeting), Roosevelt Hotel, New York, N. Y.

28 to Dec. 1 American Society of Refrigerating Engineers (50th Annual Meeting), Ben Franklin Hotel, Philadelphia, Pa.

28 to Dec. 3 American Society of Mechanical Engineers (annual meeting), Statler Hotel, New York, N. Y.

APRIL, 1955

13-15 American Society of Lubrication Engineers (10th Annual Meeting & Exhibit), Hotel Sherman, Chicago, Ill.

MAY, 1955

14-22 ACHEMA XI, Chemical Apparatus & Equipment Congress & Exhibition, Frankfurt am Main, Europe.

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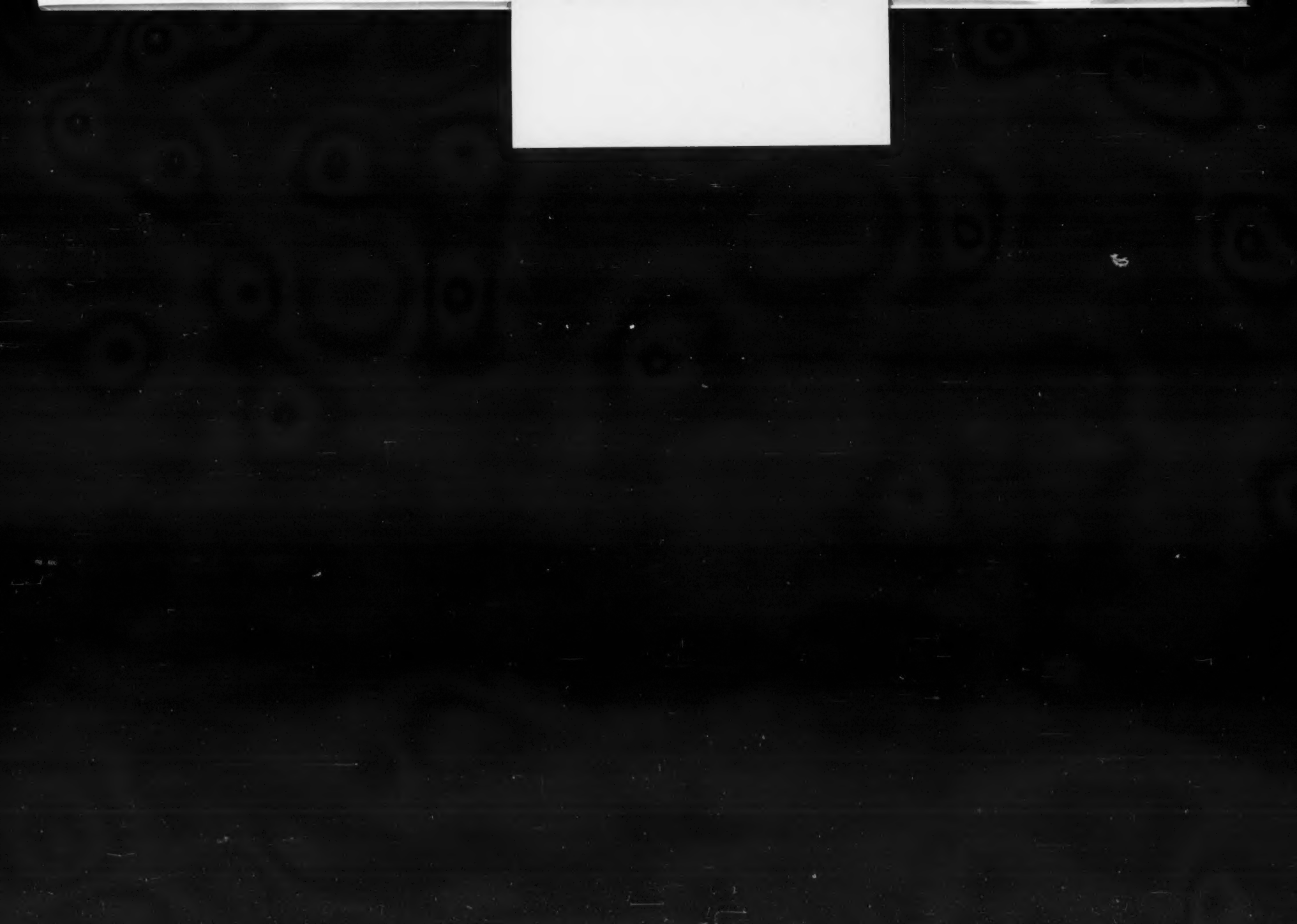


How Farval lubrication saves labor, eliminates bearing troubles on power shovels

FARVAL—
Studies in Centralized Lubrication No. 156

WATCH a power shovel tear into rock, slate, gravel, or dirt. It's dig, hoist, swing, dump—over and over again, hour after hour, day after day. Work like this soon ruins improperly lubricated bearings. That's why it pays to install the Farval Centralized System of Lubrication on quarry and mine shovels like the one shown.

It takes just a minute or two, even on a big six-yard shovel like the Bucyrus-Erie 150-B, for Farval to lubricate every bearing needing protection for heavy duty. Compare this to stopping the shovel for 15 to 20 minutes each time the operator needs to lubricate by hand—and then he has to climb around looking for grease fittings.

FARVAL FIGHTS FRICTION

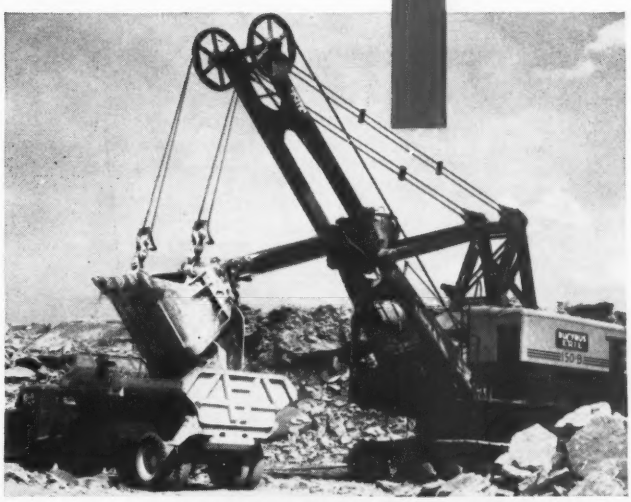
Farval delivers any oil or grease that can be pumped under pressure to any number of bearings from one central station, in exact quantities, as often as desired. The rugged Farval valve has only two moving parts—is simple and sure. No uncertain springs, ball-checks or pinhole ports to cause trouble. Any grit, abrasive sand or dust that has penetrated the bearings is forced out by the measured injections of lubricant, and the lubricant forms a protective seal. Hazards due to hit-and-miss hand lubrication are eliminated.

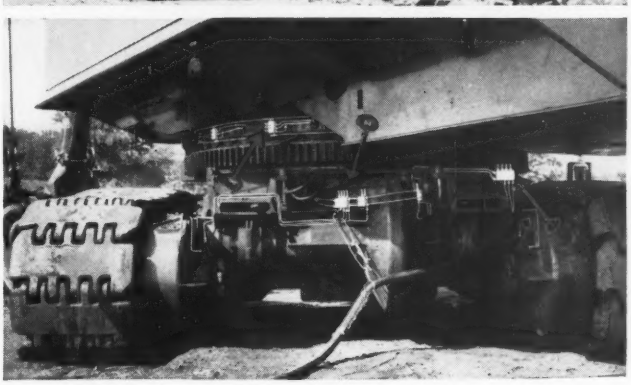
FARVAL SAVES MAN HOURS, LUBRICANT

Experience shows that Farval saves as much as 45 minutes of each hour of oiling labor—up to 3 of every 4 pounds of lubricant consumed by other methods.

If you make or use power shovels, conveyors, crushers, shaker screens, elevators or other heavy equipment, your machines will run longer and produce more if you install Farval. Farval protects millions of bearings throughout industry. Write now for Bulletin 26. The Farval Corporation, 3267 East 80th Street, Cleveland 4, Ohio.

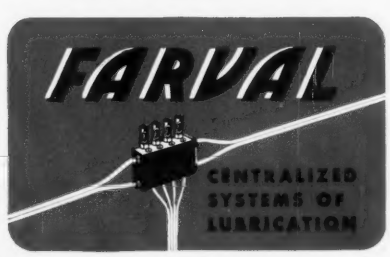
Affiliate of The Cleveland Worm & Gear Company, Industrial Worm Gearing. In Canada: Peacock Brothers Limited.





KEYS TO ADEQUATE LUBRICATION—Wherever you see the sign of Farval—the familiar valve manifolds, dual lubricant lines and central pumping station—you know a machine will be properly lubricated. Farval manually operated and automatic systems protect millions of industrial bearings.

Pictured is the Farval-lubricated truck frame of a Model 150-B six cubic yard power shovel. Photo courtesy of Bucyrus-Erie Company.



DON'T SCRAP THAT GEAR

MILL-PINION or REDUCTION

Until you first consider HODSON LUBRICATION. Gears both modern and old types, badly damaged by unsuitable lubricants, have been saved, and their usefulness extended with the "Four Horsemen" of lubrication.®

Among the many plants operating units of modern, intermediate or old type design, served with DENSOIL, GEARKOTE, METALICOIL, and METALICGREASE, material reductions have resulted in lubrication costs—less down time, less costly repairs and less parts replacement.

In most instances of any period design, both necks and teeth—gears and bearings of all types, have been or are being handled with one lubricant the year round, both Winter and Summer.

This covers Strip Mill Pinions, Blooming Mill Pinions, Structural Mill Pinions, Rail Mill Pinions, Plate Mill Pinions, combined Reduction Gear and Pinion Drives, Worm Gears, Pipe Mill Drives, Edging Mill Drives, and Reduction Gears of all types and sizes.

Preventive measures are preferable to corrective. Units similar to the above, containing the original installation of HODSON Lubricants, have been operating for 25 to 30 years, without removing a cap or cover bolt.

May we offer you the benefit of our personal practical experience and services as consultants for your lubrication problems without obligation or charge.

PRODUCT NAMES GEARKOTE, METALICOIL AND METALICGREASE ARE REGISTERED TRADE MARKS OF THE HODSON CORPORATION

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Lubrication Engineers and Manufacturers

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